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(English Edition)

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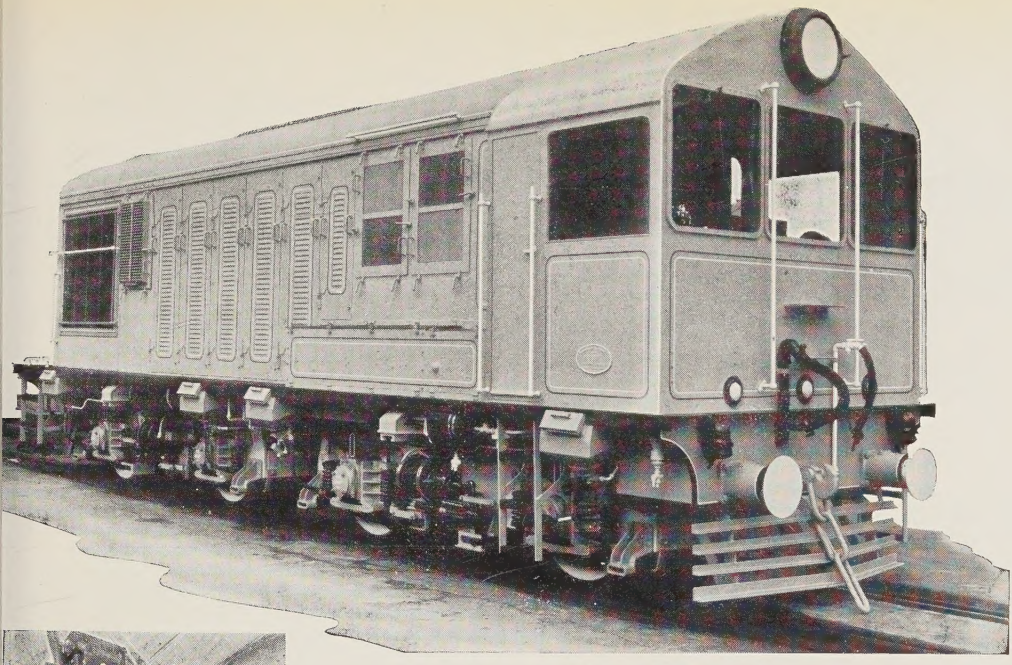

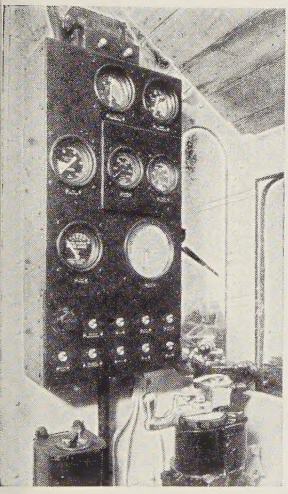

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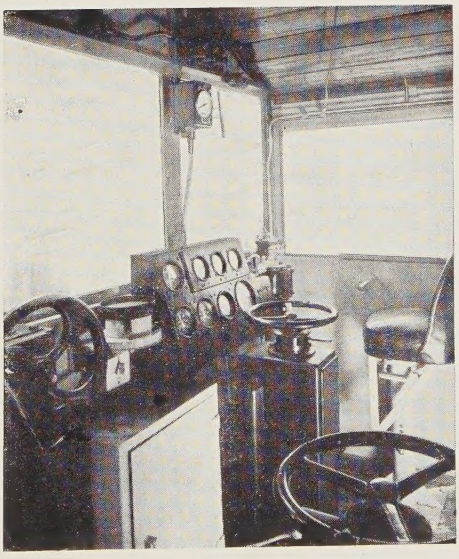

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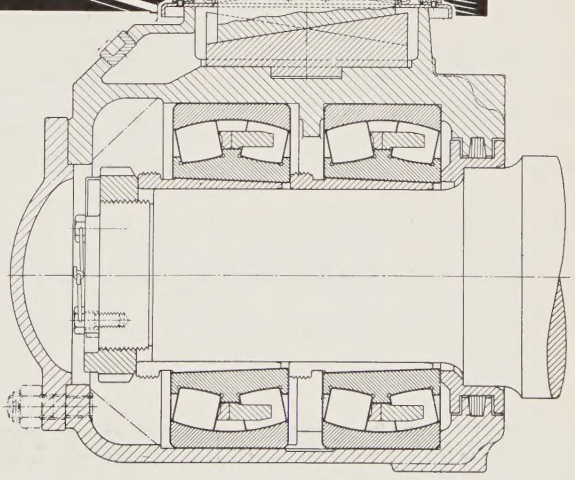

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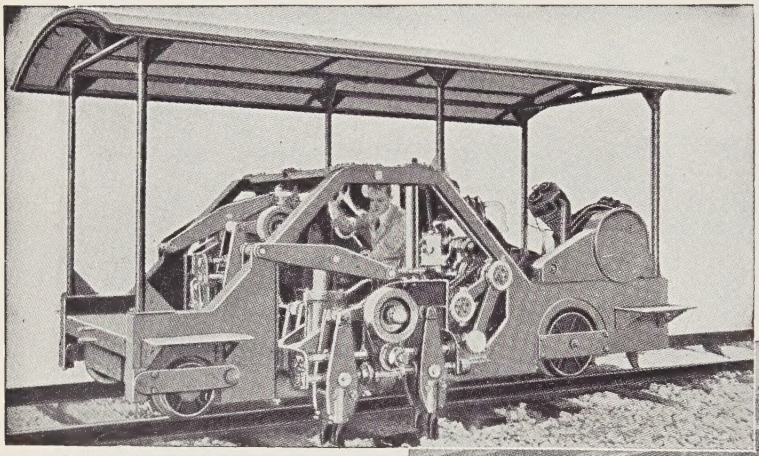

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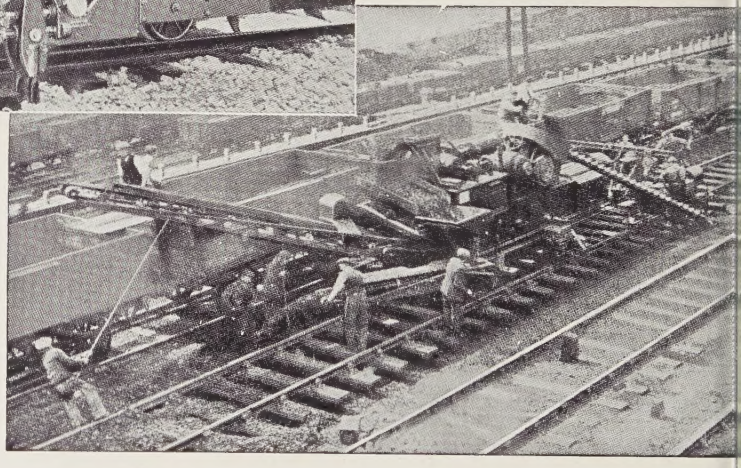

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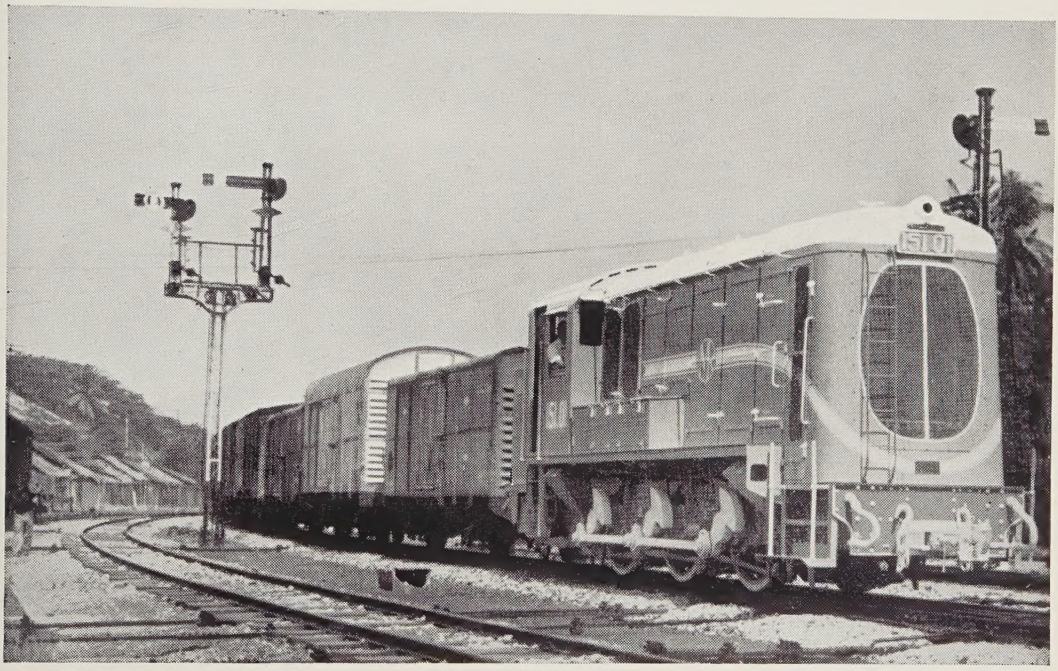

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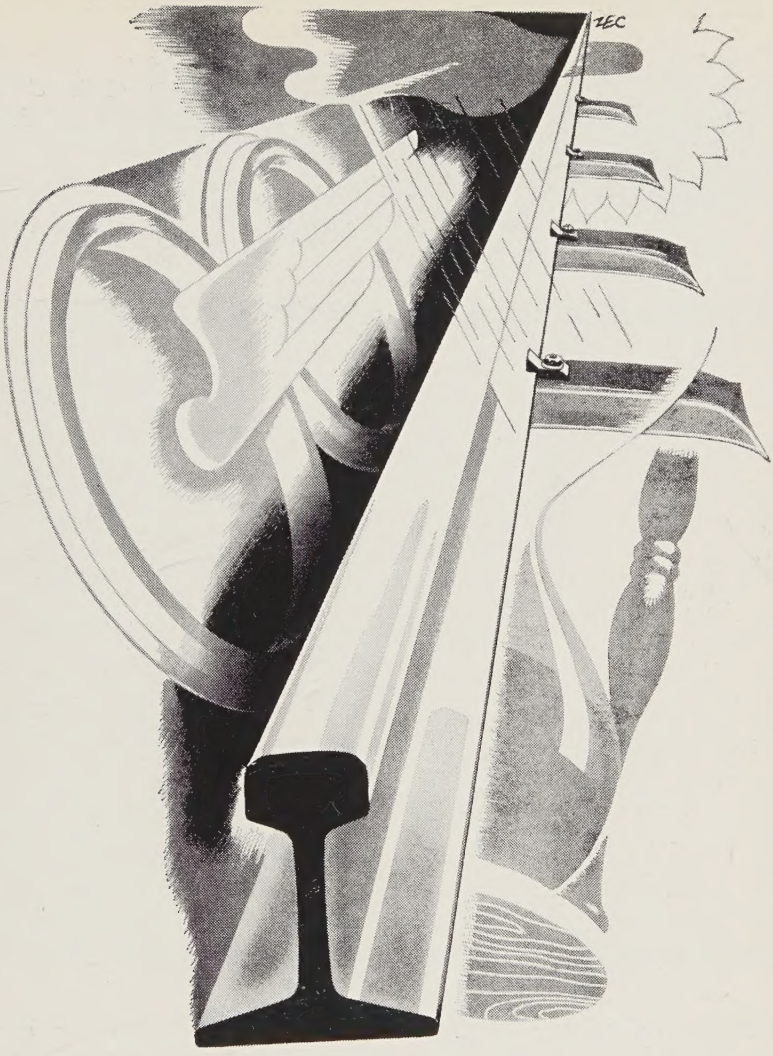

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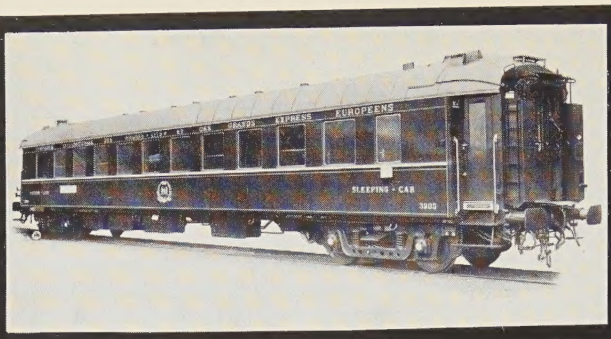

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Vol. XXVIII. — No. 5. May 1951.

BULLETIN
OF THE
INTERNATIONAL RAILWAY CONGRESS
ASSOCIATION
(ENGLISH EDITION)

[625 .14 (01 & 625 .2 (01]

Wheel and Rail,

by P. LABRIJN, (†)

Former pupil of Delft Polytechnic, Assistant Chief Engineer Locomotive and Rolling Stock Department, Netherlands Railways.

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Model arrangement for determining transverse force likely to cause derailment of an axle.
Full scale trial with three-axle tender to determine transverse derailment force.
Model arrangement for determining risk of derailment at high speeds.

INTRODUCTION.

In 1926, the Locomotive Commission of the German Railway Union (Verein Deutscher Eisenbahnverwaltung) (since 1933, the « Central European Railway Union ») decided to form, within its organisation, a Commission for the revision of paragraphs 87 and 88 of the Technical Conventions (Technische Vereinbarungen) in force at the time.

The Commission comprised representatives of the Reichsbahn Karlsruhe Direction (President), the Reichsbahn General Management at Berlin, the Reichsbahn Stuttgart Direction, the Austrian Federal General Management at Vienna and the Netherlands Railways, Utrecht.

In 1930, the name of the Commission was modified to *Sub-Committee for investigating the running of locomotives.* In 1931, representatives of the Way and Works Commission of the Union were

added to the Sub-Committee, these being from the Berlin Reichsbahn General Management and the Austrian Federal Railways, Vienna.

The Commission was charged, firstly with the study of problems of locomotive running over points and curves, particularly with a view to safe and smooth running.

The good behaviour of a locomotive and ease of running over curves and turnouts is of extreme importance. It covers the qualities which assure safety.

The last meeting of the Commission was in May 1939. As we do not know whether the uncompleted work of this Commission was afterwards proceeded with, it does not appear inappropriate to furnish here some details of the different questions studied, with the aim of providing a starting point, if it should be required, for further study. We shall deal particularly with the part taken by the Netherlands Railways in this work. Most of the trials were carried out in the Works of the Netherlands Railways.

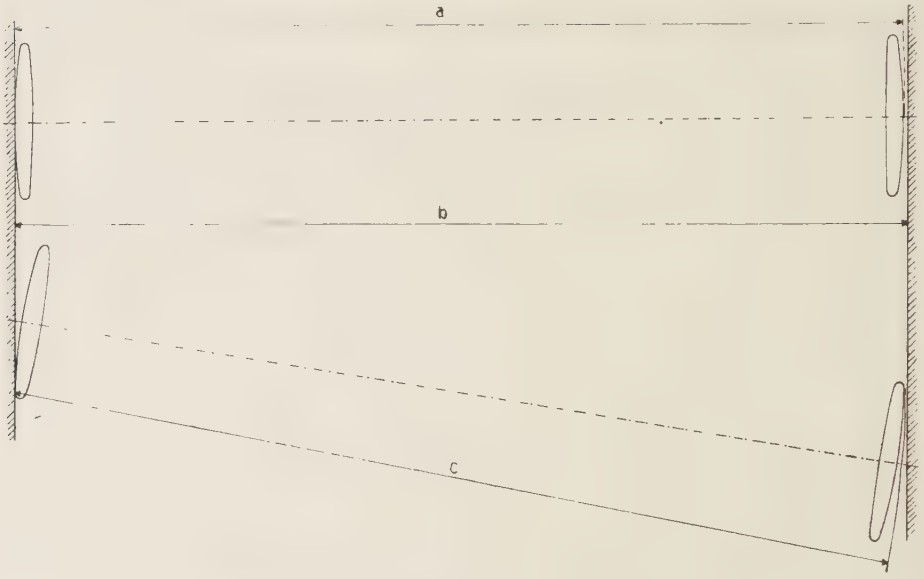


Fig. 1.

a = Maximum distance between flanges at 10 mm (3/8 in.) below the running surfaces = ± 1426 mm (56.14 in.).
b = Minimum gauge of track at 10 mm below the running surfaces of the rails = ± 1436 mm (56.53 in.).
c = Angular measurement of flanges of 1 m (39.37 in.) dia. wheels, 10 mm outside the running surfaces = ± 1430 mm (56.29 in.).

General remarks.

For a vehicle to negociate a given curve, it is necessary that it can be inscribed geometrically in the curve.

A two-axled vehicle can be geometrically inscribed in any curve, since a single pair of wheels on the track can be turned

around its vertical axis without the flanges bearing on the inner faces of the two rails. This is possible because of the transverse play of at least 10 mm (3/8″) which exists between the edges of the flanges and the inner faces of the two rails (see fig. 1).

The running of a two-axled vehicle in a curve is at the same time limited by the angle at which the flange of the leading axle bears on the outer rail. The angle should not exceed a certain maximum value.

If it is specified in the case of a two-axled vehicle that the rear axle should be capable of radial inscription and in contact with the inner rail, the position of the vehicle is shown in fig. 2, in which :

A = leading axle;

B = trailing axle;

p = transverse play;

a = wheelbase of the vehicle;

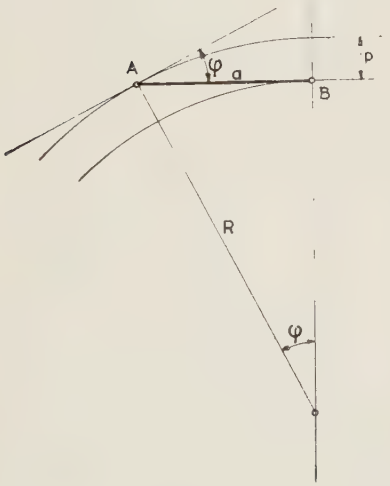


Fig. 2.

φ = bearing angle of leading axle;

R = radius of curve (outer rail).

From this diagram we have :

$$2p\mathrm{R} = a^2$$

$$a = \sqrt{2p\mathrm{R}}$$

$$\sin \varphi = \frac{a}{\mathrm{R}}$$

Table I summarises the various values of the above for certain radii.

We thus see, for example, that in the case of a 300 m (984 ft.) radius curve with 25 mm (1 in.) widening of the gauge and a vehicle with 4.59 m (14 ft.) wheelbase, the rear axle would be in exact lateral contact with the inner rail in the radial position and that the bearing angle of the leading axle is 0°53′. In the case of a smaller wheelbase or greater radius of track curvature, the rear axle would not touch the inner rail.

Vehicles with more than two axles, particularly if their wheelbase is rather longer, cannot conform to any curve whatever. To facilitate their inscription in curves, there are various methods, such as : slackening of the gauge; reducing the flanges of intermediate axles; transverse play of several axles; use of articulation for an axle or bogie.

To determine the degree to which a vehicle with more than two axles can inscribe a curve and the bearing angle of the guiding axle, the graphic processes of ROY (1) and VOGEL (2) can be used. The ROY procedure is an approximation

(1) See : *Revue Générale des Chemins de fer*, 1884, p. 153.

(2) See : *Organ f. d. Fortschritte des Eisenbahnwesens*, 1926, p. 354.

TABLE I.

R	Slackening of the gauge	p	a	sin φ	φ
m	m	m	m	—	—
180	0.025	0.035	3.55	0.0197	1° 8′
210	0.025	0.035	3.84	0.0183	1° 3′
250	0.025	0.035	4.19	0.0167	0° 58′
300	0.025	0.035	4.59	0.0153	0° 53′
400	0.015	0.025	4.49	0.0112	0° 38′
500	0.015	0.025	5.00	0.0100	0° 35′

and cannot be used with sufficient accuracy for vehicles of limited length on curves of large radius, whilst the VOGEL procedure is also for fairly long vehicles on sharp curves.

In the question posed, it is necessary firstly to determine the forces which act between rail and wheel during the passage of a vehicle through a curve, it being assumed that the risk of derailment is the consequence.

When we consider the movement of a locomotive or other vehicle through a curve, it is obvious that the direction of movement of the wheels at the points of contact with the rails changes continually. The movement can be considered as the pivoting of the vehicle around a certain point, which moves with the vehicle in the direction of running.

When passing through a curve, one or more flanges bear laterally against the rails, which gives the guiding force necessary to keep the vehicle on the track and to cause it to pivot around the point mentioned above. During this pivoting movement, the various wheels in movement slide simultaneously on the rails, generally in a longitudinal or transverse direction. The frictional forces are consequently opposing, and must be overcome by the guiding forces of the flanges which bear on the rails laterally. The problem thus comprises in the first place the determination of the point around which the vehicle appears to pivot during its passage through a curve.

Forces between wheel and rail when running through curves.

When a wheel takes a position with the flange bearing against the rail, we note between wheel and rail, the vertical load Q of the wheel at the point of contact with the running surface of the rail, and the angle of thrust Y at the point of contact A of the flange (see fig. 3). For the wheel to become derailed, it must be lifted

until the flange surmounts the rail. The first effect of lifting the wheel would be the disappearance of the point of contact with the running surface and there would remain only the point of contact A of the

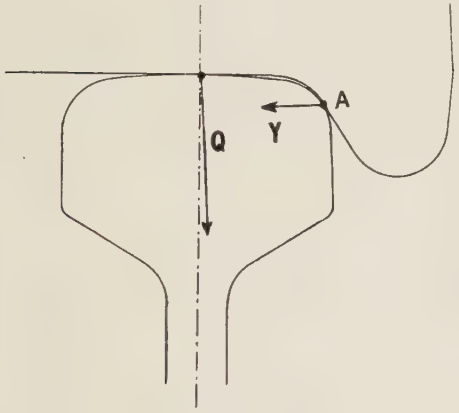


Fig. 3.

flange (see fig. 4). From this time, the wheel would turn on the latter point. The vertical load of the wheel would in this case also operate on point A.

The bearing wheel has, therefore, a movement consisting of running and

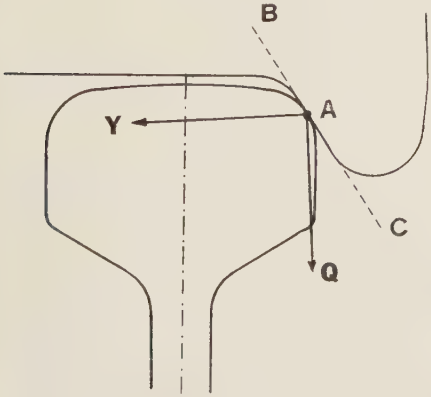


Fig. 4.

sliding on the upper surface of the wheel and continual sliding downwards on the side wall of the rail with increasing bearing pressure at some points on the running surface of the rail until the sum of the forces operating in the direction of the inclined face of the flange (direction BC) upwards, exceeds the frictional forces at the point of contact.

Thrust of flange on rail.

If the horizontal bearing pressure at the point A (3) — which we shall call flange thrust — is Y (see fig. 5), the force Y can be separated according to the inclined plan BC of the flange face and the perpendicular direction of it. These factors have values Y cos α and Y sin α. The vertical load Q of the wheel, facterised in the same way, gives factors Q sin α and Q cos α.

The resultant force acting on point A in the direction BC is :

$$R = Q \sin \alpha - Y \cos \alpha,$$

whilst in the direction CB, is the frictional force :

$$F = f(Y \sin \alpha + Q \cos \alpha),$$

when the force R is equal to the force F. In this case the wheel finds a bearing on point A and commences to run on this point and to lift itself.

In this extreme case, we get the equation :

$$Q \sin \alpha - Y \cos \alpha = f(Y \sin \alpha + Q \cos \alpha)$$

(3) Although point A may not be exactly in the vertical axis of the wheel, this fact will be ignored later for the purposes of the calculation.

or :

$$Y = Q \frac{\sin \alpha - f \cos \alpha}{\cos \alpha + f \sin \alpha} = Q \frac{tg\ \alpha - f}{1 + f\ tg\ \alpha} \quad (4)$$

Making $f = \text{tg}\ \varphi$, we get :

$$\frac{Y}{Q} = tg\ (\alpha - \varphi).$$

If we introduce the value $f = 0{,}20$, 0.25 and 0.35, or $\varphi = 14^{o}$, we get for the various inclinations of the face of the flange, the values given in Table II.

Direction of movement.

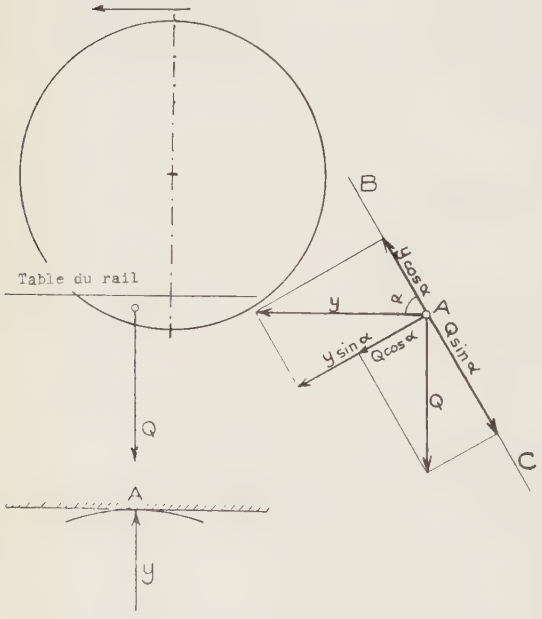


Fig. 5.

Explanation of French terms :

Table du rail = Running surface.

(4) This formula was established by MARIÉ, although in another manner. See : *Traité de stabilité du matériel des Chemins de fer*, Georges Marié, Paris and Liège, 1924.

TABLE II.

α	Y : Q for		
	$f = 0.20$	$f = 0.25$	$f = 0.35$
30°	0.338	0.287	0.188
40°	0.547	0.488	0.353
50°	0.800	0.727	0.593
60°	1.14	1.04	0.859
70°	1.64	1.48	1.22
80°	2.56	2.25	1.78
90°	4.99	4.01	2.85

The values Y : Q for $f = 0.25$ (up to the maximum value of 2) are traced in figure 6.

If the point A is located on the straight position of the flange profile, between the root and the nose, which arises when the wheel begins to lift, in the case of unstable balance the angle has a value of 60°, i. e. with a coefficient of friction $f = 0.25$ and a flange thrust equal to the approximate value of the wheel load, the wheel bearing on the rail will lift and become derailed.

It may be remarked, however, that at the same time as the bearing wheel begins to climb the rail, it is necessary to consider the friction of the other wheel of the same axle on the other rail. If the position of the axle in relation to the track is oblique to the point where the friction of this wheel also tends to cause the bearing wheel to mount the rail, then this friction is active and not passive. The load of this wheel

n the rail at the moment in question eing Q and the coefficient of friction f', e shall have $Y_1 - Y - Qf'$.

It must be noted that Q does not epresent the static load, but the actual oad at the moment of derailment. It may e taken into account that by one of the vheels bearing on the rail, this wheel is verloaded whilst the load on the other vheel of the same axle is relieved.

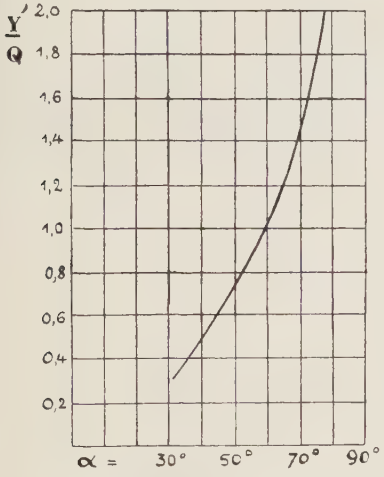


Fig. 6.

The force Y_1 of the bearing wheel herefore causes this wheel to mount the ail, whilst the friction of the other wheel avours the derailment. The unloading of his wheel has an advantageous effect, s the amount of friction is reduced, hilst, in addition, the bearing wheel is the same time overloaded, which pposes the derailment.

An experimental trial has confirmed the bove thesis.

Model arrangement for demonstrating the action of static forces between wheel and rail.

To determine the static forces set up between a wheel which runs against the rail at an angle of bearing we will consider a vehicle as in figure 7. This vehicle is fitted with an axle A, arranged at a certain angle to the centre line of the vehicle, on which is a wheel W with a flange. The axle has a small amount of play in the brasses in a longitudinal direction. The wheel W runs on a third rail with a load W.

By moving the vehicle in the direction of the arrow, the wheel W will run on the third rail in the plane of this wheel until the flange bears on the rail. From thence the wheel will tend to continue running in the same direction, whilst this will be prevented by the flange, so that during the rotation there will at the same time be produced a sliding of the wheel on the rail in the transverse direction of the latter, which will give rise to a frictional effort Qf' at the point of contact with the running surface of the rail, given that f' is the coefficient of friction of a running wheel, which is at the same time sliding transversely. This force Qf' is cancelled by a reactive force Y of equal value between the wheel flange and the side wall of the rail. The two forces Qf' and Y are not identical, since they do not bear on the same point. The bearing point of the flange, where the force Y is applied, is a little below the friction force Qf' on the upper surface of the rail and also oblique to and forward of the vertical plane of the wheel meridian in the direction of movement (recovery) by reason of the

oblique position of the axle in relation to the rail.

Given that the angle of bearing is in practice fairly small, the minor differences mentioned above are negligible, so that we can, without appreciable error, assume that the two forces Qf' and Y are identical. There is therefore no outside force operating on the rail and the latter could be detached from its supports.

Let us now consider a vehicle arranged as in figure 8. Here the axle A is prolonged at one end, threaded and fitted with a nut M. By means of this nut, the axle can be moved longitudinally and in consequence the inserted spring V more or less compressed. The flange of the wheel W will therefore be drawn with a certain force against the rail R, whilst the normal carrying wheels of the vehicle will be pushed against the left-hand rail.

Figure 9 represents the forces which result. We will note that the exterior forces operating on the vehicles are : between the wheel R and the rail N, the flange pressure Y and the friction effort Qf' and between the wheels W_1 and W_2 and the rail on which they run, the flange, thrusts F_1 and F_2.

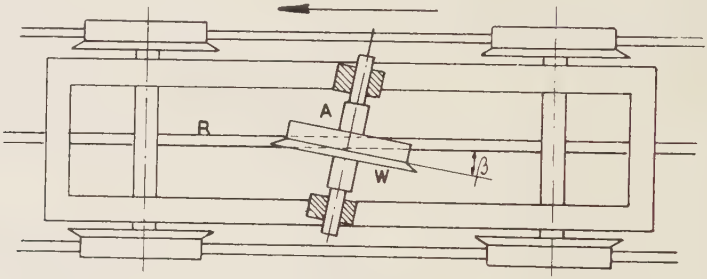


Fig. 7.

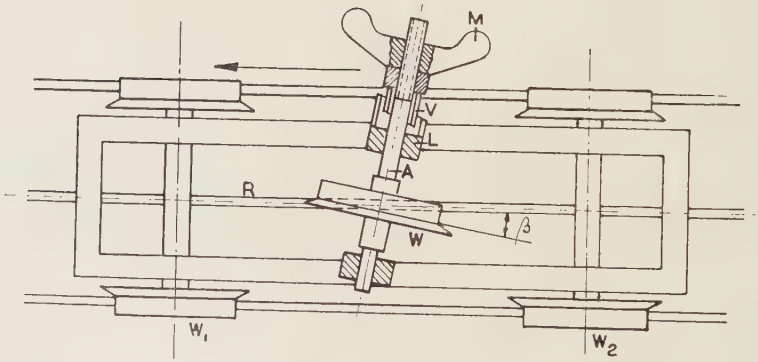


Fig. 8.

The tension force K of the spring V is an internal force.

So far, we have :

$$K = Y - Qf' \text{ and } Y - Qf' = F_1 + F_2.$$

In this case the external force operating on the centre rail is not nil.

Assuming only that forces K, Y and Qf' operate perpendicularly on the track, the accuracy obtained is quite satisfactory.

High speed running on non-continuous rails.

Let us consider the case where a wheel thrusts at a speed V with an angle of bearing β against a rail having a change of level (see fig. 10). Shock arises at a speed V sin β perpendicularly to the rail and at a speed V cos β along the rail.

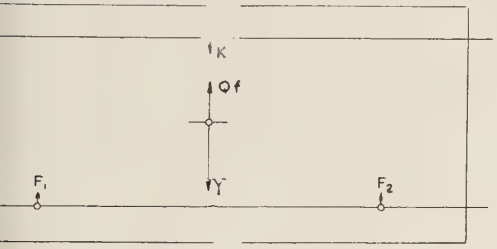


Fig. 9.

In a direction perpendicular to the rail, the kinetic energy has a value :

$$1/2\ m\ V^2 \sin^2 \beta,$$

if m is the part of the mass of the vehicle taken into account for shock, which we shall call the « reduced mass ». To determine this, it must be noted that the kinetic energy is absorbed by the deformation of the wheel and the rail, by the movement of the vehicle around its centre of oscillation and by the resilience of the bearing springs. If the road traversed by the shock force perpendicular to the rail is s [5] and if this force increases in an approximately linear manner from 0

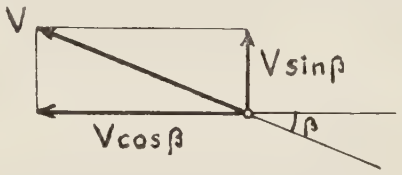


Fig. 10.

up to a value P, the average amount of this force will be 1/2 P. The potential energy developed by the shock force is therefore 1/2 Ps, which provides the equation :

$$1/2\ m\ V^2 \sin^2 \beta = 1/2\ Ps$$

or :

$$P = \frac{m\ V^2 \sin^2 \beta}{s}$$

It is not possible to resolve the value of P, since the values of m and s are not known. It only remains therefore to resolve the value of P by taking the assumed values of m and s.

Taking $\alpha = 60^o$ (see fig. 6) and $f = 0.25$ we shall have $Y \approx P$.

For example with Q = 6 000 kg, Y = 6 000 kg. Assuming finally that m = 1 200 kg sec^{2}m^{-1}, we deduce from the formula mentioned above :

$$6\,000 = \frac{1\,200\ V^2 \sin^2 \beta}{s}$$

(5) We will consider the value of s as a fictitious one since this takes into account all the displacements and distortions quoted above.

or

$$\sin^2 \beta = \frac{6\,000\ s}{1200\ V^2}$$

or, further :

$$\sin \beta = 2.236 \frac{\sqrt{s}}{V}$$

where s is expressed in metres and V in km/h.

The relation between V and s is shown graphically in fig. 11. The higher points of the curves signify danger of derailment and the lower points safe running.

On the Netherlands Railways, a maximum angle of bearing of 2° is permitted, which has not shown any disadvantages; we can conclude that, if the running speed is 45 km (28 miles)/h, the value of s can be at least 40 mm (1 9/16″). Assuming that this is apt, a bearing angle of 50′ would be allowable at a speed of 120 km (75 miles)/h.

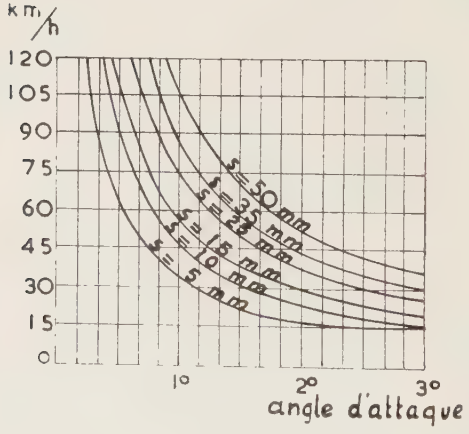


Fig. 11.

Explanation of French terms :

Angle d'attaque = Angle of bearing.

« Free » or « tight » running over curv

When a vehicle runs on a curve, t flange of the leading outer wheel bea on the inner face of the outer rail of t track, or, briefly, the leading axle bea on the outer rail; the rear axle will follo a straight line and will tend to positi itself approximately on the radius of t curve.

In figure 12, there is the possibili owing to the size of the wheelbase, transverse play of the set of wheels the track, and owing to the radius curvature of the track that the rear ax will take up a radial position. In th case the flange of the inner wheel of t rear axle does not touch the inner ra of the track, or in other words the re axle does not bear against the inner ra The position of the vehicle in the track not fixed exactly since the rear axle can moved a little transversely to the trac

Figure 13 illustrates a case where the re axle also is in contact with the inner ra This axle cannot run radially. The positi of the vehicle in the track is fixed, as t

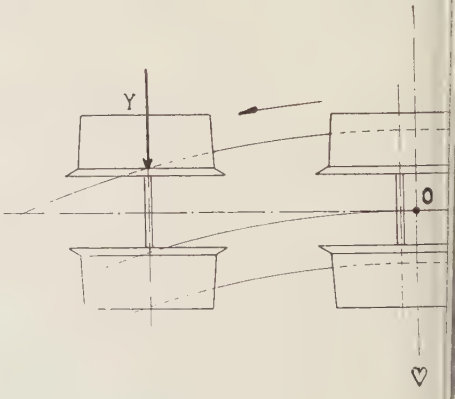


Fig. 12.

front axle bears against the outer rail and the rear axle against the inner rail. The vehicle is guided by two axles. This we shall call « tight » running; conversely the position explained earlier we shall call « free running ».

Heumann's theorem.

We see from figure 12 that only the flange of the front axle bears on the outer rail, and from figure 13 that the rear axle also bears on the inner rail. These contacts set up forces which cause the vehicle to pivot constantly during its passage on the curve and at the same time to cause the wheels whilst rotating to slide both transversely and longitudinally. We shall call the point around which the vehicle, at the time turns, the « pivot-pole of the vehicle » (6). UEBELACKER (7) was the first to take into account the importance of this pole. He calls it the centre of friction, because the various frictional efforts of the wheels on the rails find a common centre there.

According to him, the pole is, in the case of figure 12 (free running) located a little behind the rear axle on the longitudinal centre line of the vehicle; in the case of figure 13 (tight running) the location of the pole is determined by the foot of a norm, dropped from the centre of the curve on the longitudinal centre line of the vehicle. With this as a basis, it is assumed that the wheel tyres have been turned cylindrically and the wheel flanges have been replaced by rollers with vertical axes.

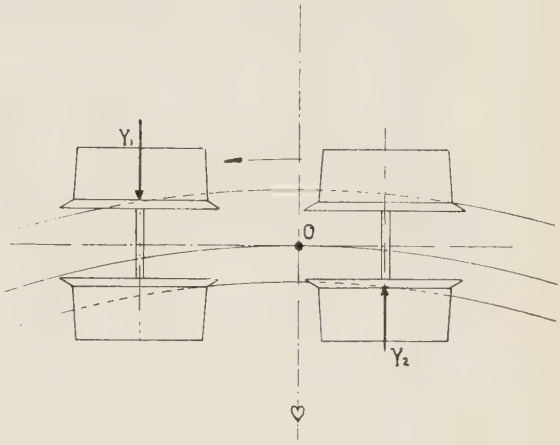


Fig. 13.

Determination of the guiding forces (flange thrusts) is important having regard to the danger of derailment of the axles under consideration.

Let us suppose that the point O is the pole (see fig. 14); in this case, the wheels of the leading axle 1 pivot on point O and exercise sliding forces K_1. The direction of the forces K_1 is perpendicular to the straight lines joining the poles and the points of contact between wheels and rails on the running surfaces of the latter.

We have : $K_1 = Qf''$

in which :

Q = wheel load.

f'' = centre of friction between wheel and rail in the case of transverse and

(6) In fact, the centre of pivoting of the vehicle when running through a curve — provided the radius does not change — is the centre of curvature of the track. The friction pole mentioned above is the common centre around which move the points of contact of wheels on the rails, providing that the load is the same on all wheels and that they have no driving effort, nor brake effort. In the latter cases, the centre of pivoting of the various wheels are not located in the same region and generally not in the axis of the vehicle.

(7) See : *Organ f. d. Fortschritte des Eisenbahnwesens*, 1903, BEILAGE.

longitudinal skidding of the wheel whilst running.

In a similar way, the skidding forces of the wheels of the rear axle II are represented by the forces K_2 in size and direction.

We then have $K_2 = Qf''$, taking the load of the rear axle as equal to that of the front axle.

To overcome the frictional efforts of the wheels on the rails during passage through a curve, it is necessary to exert in free running a pivoting moment M of the vehicle (see fig. 14) :

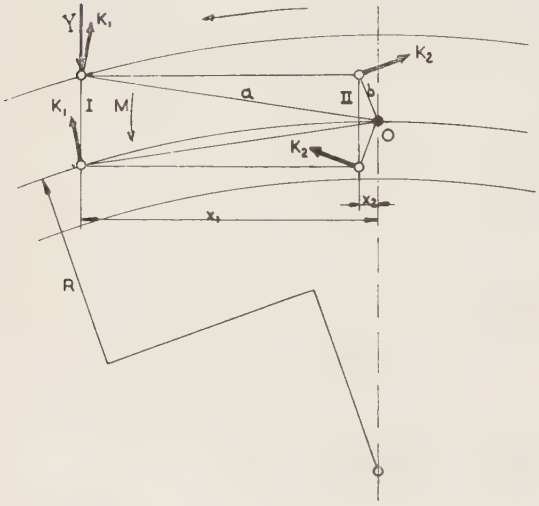


Fig. 14.

$$M = 2K_1 a + 2 K_1 b$$

or :

$$M = 2 Q f'' (a + b).$$

This moment must be provided by the guiding force Y of the outer wheel of the leading axle (the only wheel with a flange bearing on the rail).

HEUMANN developed a graphical method of determining the location of the pole on which the vehicle pivots during its passage through a curve ([8]). He referred to the fact that in theoretical science the location of the pole is such that the guiding force is a minimum one, basing this on the general thesis that all work naturally produced is according to the rule of lowest effort.

Let us now consider figure 16. The straight line AB represents the wheelbase ($= c$) of the vehicle the leading axle I being at A, the rear axle (II) at B. The straight line AB is divided into a certain number of equal parts, the division being continued beyond its point B. The straight line CD is traced parallel to AB at a distance $1/2s$ (= half the gauge). The straight line AB may be regarded as the axis of the vehicle.

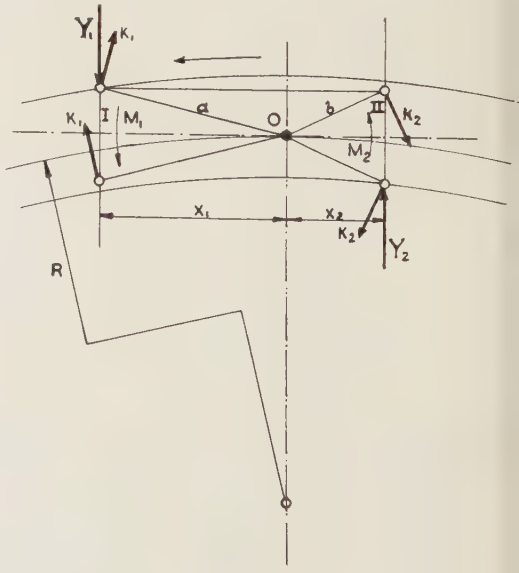


Fig. 15.

([8]) See : *Organ f. d. Fortschritte des Eisenbahnwesens*, 1913, p. 104.

At all points A, B and 1 — 6, the distances a and b are determined (drawn on fig. 16 for the point of division 4) and the sum of the distances (or $a + b$) is taken perpendicularly upwards from AB. The extremities plotted are joined to give

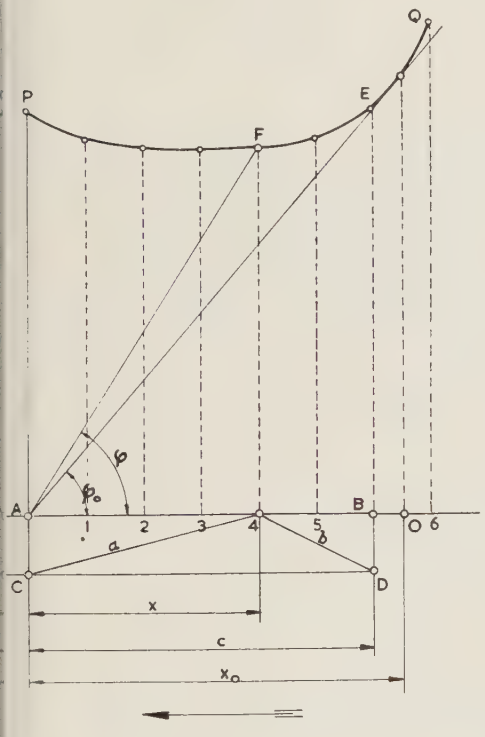


Fig. 16.

the curve PQ. The tangent of the curve to point A provides the tangent point E. If an upright is taken from point E to the line AB, the base O represents the desired pole.

Example. — The guiding force $Y = \frac{M}{x}$, in which x = the unknown distance from the guiding force to the pole.

$$\frac{M}{x} = \frac{2 Q f'' (a + b)}{x} = 2 Q f'' \frac{a + b}{x}$$

At any point, for example, at point 4 we have :

$$\text{tg } \varphi = \frac{a + b}{x}$$

then :

$$Y = 2 Q f'' \text{ tg } \varphi$$

This value is a minimum with a minimum value of tg φ and therefore with a minimum of φ.

The angle φ has a minimum value of φ_0, i. e. the value corresponding to the point O.

With a vehicle running « tight », where the trailing axle also bears on the rail, the following method is used :

The location of the pole is determined by the foot of the norm dropped from the centre of the curve on the longitudinal centre line of the vehicle (see fig. 15).

In figure 17, the curve PQ is traced identically to figure 16, assuming we are considering the same curve and the same vehicle, the axles at the same time having less transverse play in the track, so that the vehicle is tight and the rear axle cannot move radially. Given that the curve RQ represents by its ordinates the distances from the pole to the points of contact between wheel and rail, that is that these ordinates also represent for the various points the moments of the frictional forces between wheel and rail in relation to the pole, it is obvious that, if these forces are taken as equal to the unit, the curve PQ is the same in the two cases represented in figures 16 and 17. The

point O being the pole and the point E being known, we can trace the tangent GF to the curve PQ from the point E.

We know that the moment of the guiding force of the leading axle Y_1 in relation to this axle is nil, as is the moment of the guiding force of the trailing axle Y_2 in relation to the latter.

If we now trace the straight line BG, we divide the region ABFG into two parts, the upper of which represents the moments of the guiding force of the leading axle Y_1 (nil at point A) and the lower part the moments of the guiding force of the rear axle Y_2 (nil at point B).

We see that, in relation to point O, the moment of the guiding force Y_1 is M'_o and the moment of the guiding force Y_2 is M''_o from which we get :

$$Y_1 = \frac{M'_0}{x_1} = \frac{M'}{c}$$

and :

$$Y_2 = \frac{M''_0}{x_2} = \frac{M''}{c}$$

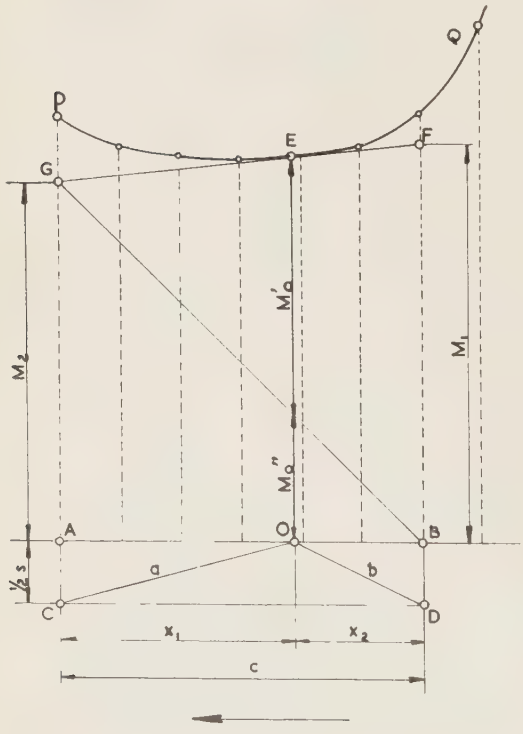


Fig. 17.

Test bed for determining the coefficient of friction between wheel and rail. — Experimental determination of areas of contact between wheel and rail.

It is proposed firstly to determine the forces which must be applied to a wheel in rotation to cause it to slide transversely. A wheel bearing on the rail at a given angle must slide with the other wheel of the same axle by the action of the guiding force of the bearing wheel. The resistance to this sliding movement must be cancelled by the guiding force. The coefficient of friction f' of a wheel in rotation, displaced at the same time transversely was not, so far as we know, determined at the time of our trials (9). This coefficient is frequently assumed to be 0.25 which is fairly high. It seems possible that the coefficient f' is less than the sliding coefficient f when at rest, given that a calculation with coefficient f would give guiding forces,

(9) It was nevertheless known that HAMELINK had carried out simple tests with a two axle test vehicle with a total weight of 7.1 kg from which he concluded that the coefficient of friction in the case of transverse displacement of a wheel in rotation is equal to the coefficient of friction at rest, viz. 0.25. See : *De Ingenieur*, 1918, p. 467.

On the other hand, we did not then know that BUCHLI had already made tests to determine the coefficient of friction in the case of transverse displacement of a rotating disc. See : *Schweizerische Bauzeitung*, 1923, p. 119.

which would be unreasonably high. If the guiding forces had a sufficiently high value to make the vehicle pivot when running through curves, it would be in a state of unbalanced equilibrium, which does not correspond with practical experience. Furthermore, this interpretation has the inconsistency that a steam locomotive on a curve tends to skid, which

flanges, were superposed (see fig. 18). The lower set of wheels represents the rail, which is not in fact correct since the running of a wheel on a rail is the running of a circle on a straight line, whilst on the test bed the movement was between two circles.

According to Hertz the maximum value of bearing pressure between two

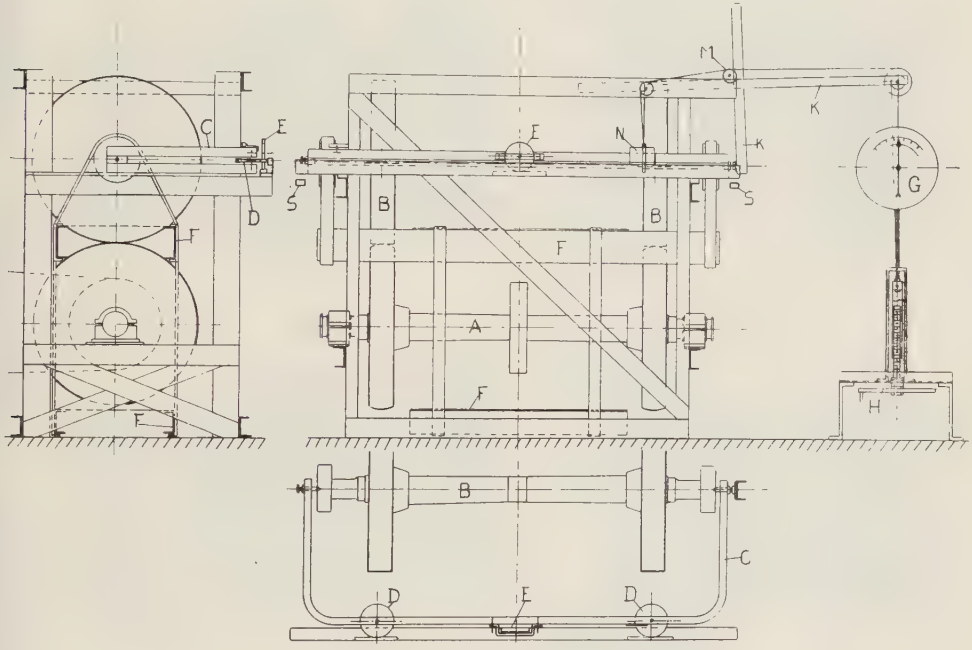


Fig. 18.

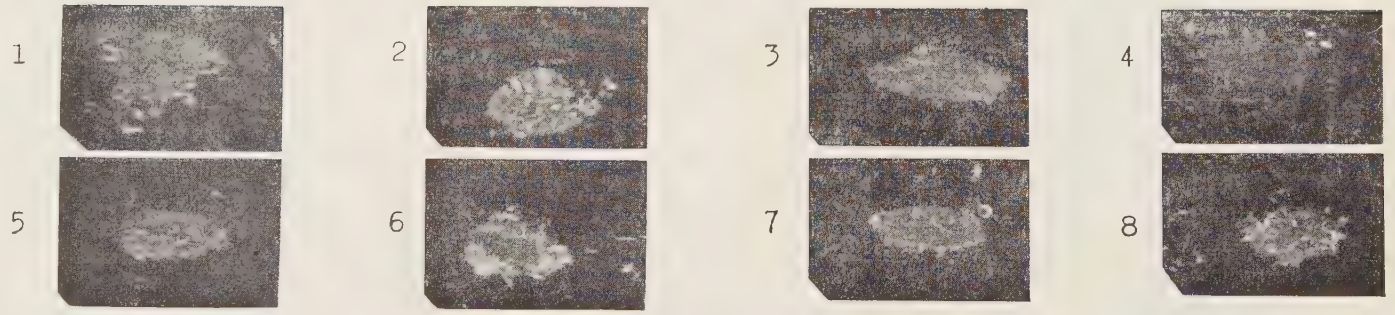
should be attributable to transverse sliding of the driving wheels.

In view of these considerations, it was decided to determine experimentally the coefficient of friction of a wheel in rotation, which is at the same time subject to transverse displacement. For this purpose, a test bed was constructed on which two sets of wheels, full size and without

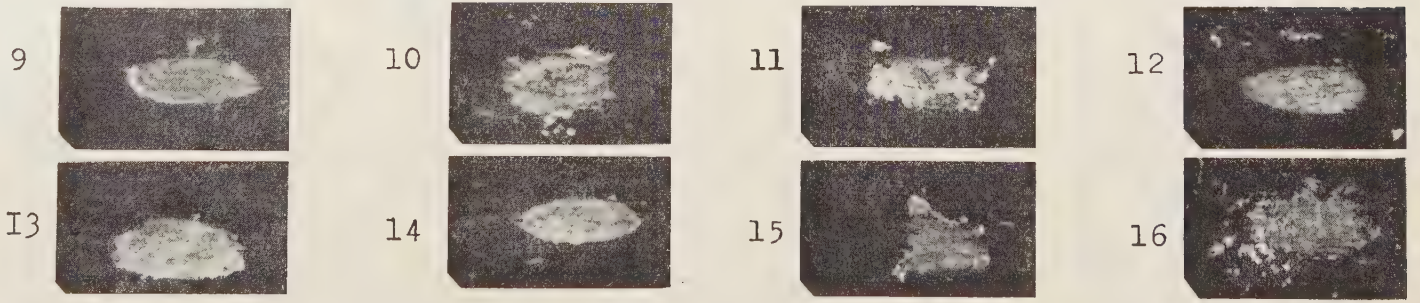
cylindrical rollers, running one on the other, has a value equal to $\sqrt{2}$-times that of a cylindrical roller running on a flat plane.

To determine the areas in contact between wheel and rail, trials were carried out by inserting very thin carbon paper. Copies from these gave clear pictures of the areas of contact (see fig. 19, items

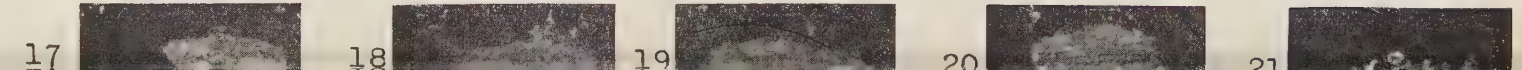
Wheel diameter 930 mm (36.60 in.).



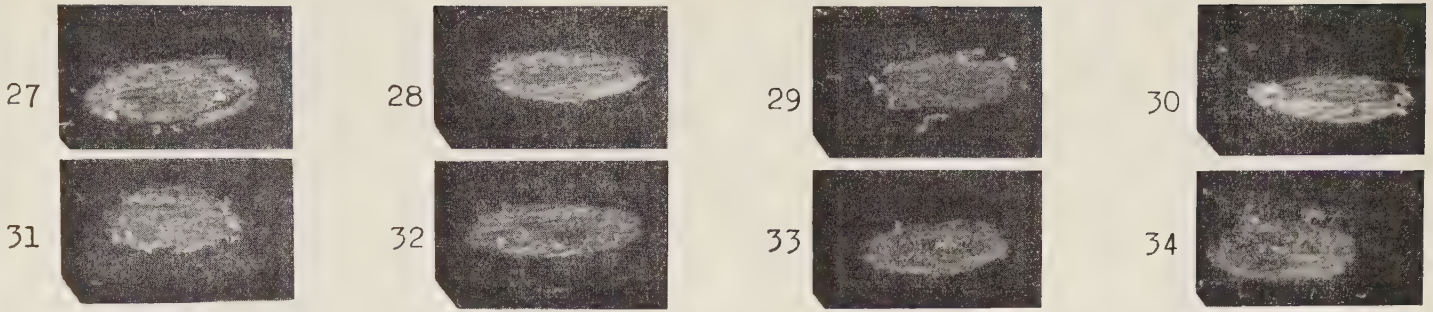
Wheel diameter 1 100 mm (43.30 in.).



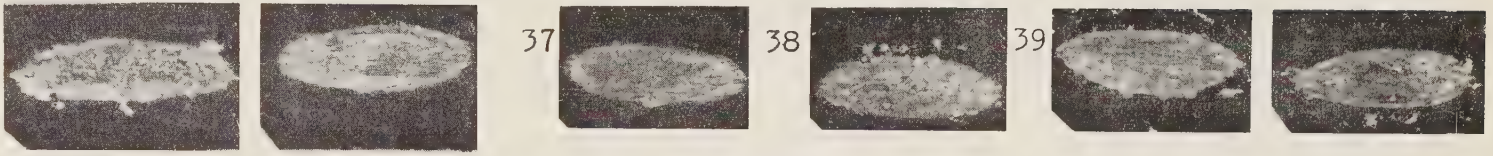
Wheel diameter 1 240 mm (48.81 in.).



Wheel diameter 1 400 mm (55.11 in.).



Wheel diameter 1 850 mm (72.83 in.).



Wheel diameter 2 150 mm (84.64 in.).

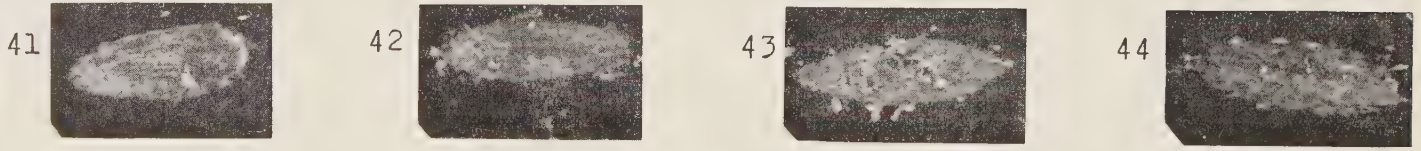


Fig. 19.

TABLE III.

Figure	Wheel dia.	Wheel load	Area of contact Surface	Pressure
—	mm	t	mm²	kg/mm²
19 (1-8)	930	5.5	100	55
		6.4	80	79
		5.8	100	58
		5.8	87	67
		6.4	80	82
		6.6	94	70
		6.2	85	73
		6.0	80	75
Average		6.1	88	70
19 (9-16)	1 100	4.6	90	51
		4.8	100	48
		4.2	100	42
		3.9	80	49
		3.3	92	36
		3.2	67	48
		3.2	80	40
		3.4	70	48
Average		3.8	85	45
19 (17-26)	1 240	4.4	100	44
		4.1	110	37
		5.1	132	39
		5.2	100	52
		2.4	80	31
		3.9	78	50
		2.9	82	36
Average		4.0	98	41
19 (27-34)	1 400	8.6	130	66
		6.6	120	55
		7.9	120	66
		6.8	112	60
		6.9	120	58
		6.7	120	56
		6.7	120	56
		6.9	123	56
Average		7.1	120	60
19 (35-40)	1 850	8.6	200	43
		8.0	160	50
		8.2	155	53
		8.3	160	52
		8.0	173	46
		8.0	148	54
Average		8.2	166	50
19 (41-44)	2 150	7.7	220	35
		7.6	220	35
		6.5	200	32
		6.8	210	33
Average		7.2	213	34

1-44). The results of the trials are shown in Table III.

The test bed comprised a set of wheels A which had the flanges removed and the tread rounded off with a 200 mm (7 7/8 in.) radius, similar to the radius of the tread

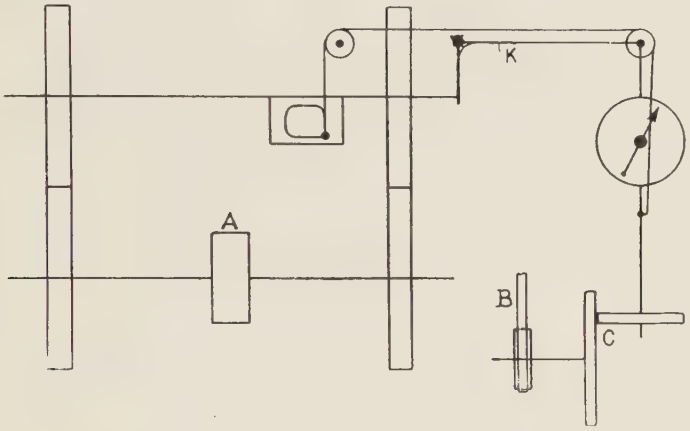


Fig. 20.

surface when new. This set of wheels rested on normal bearings and on the centre of the axle was mounted a driving pulley. The bearings were shorter than the journals so as to allow a limited longitudinal movement.

Above the lower set of wheels A was placed a set of wheels B, also without flanges and with cylindrical tread. The latter set of wheels was held in position by a yoke C, which allowed the wheels to slide in the direction of the axle. To reduce as far as possible the friction during lateral displacement, the yoke C was provided with two discs D and E. On the set of wheels B was suspended through roller bearing boxes, a ballast box F containing the loading weights for the upper set. Pressure on the contact areas between the two sets of wheels could be modified by means of these weights.

Displacement of the upper set of wheels was by a dynamometer spring which could be tensioned by a ring screw H. The thrust of the spring on the upper wheels was

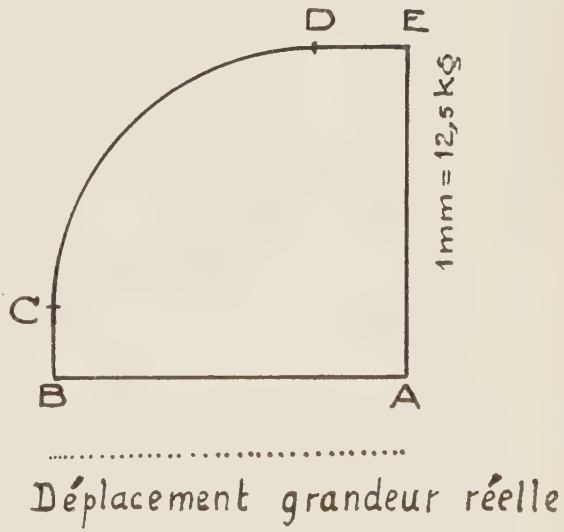
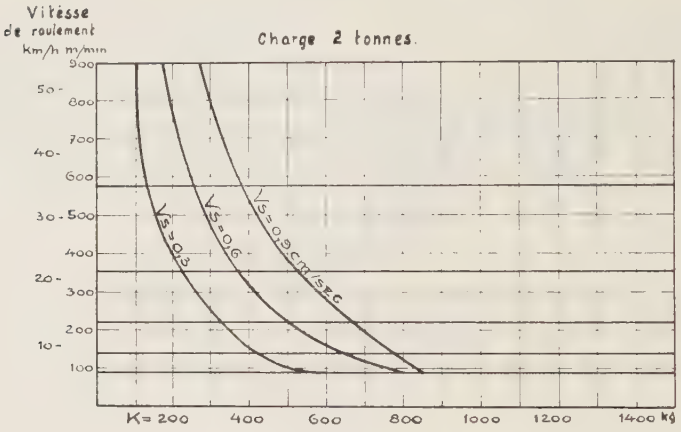


Fig. 21.

Explanation of French terms :

Déplacement grandeur réelle = Actual size displacement.

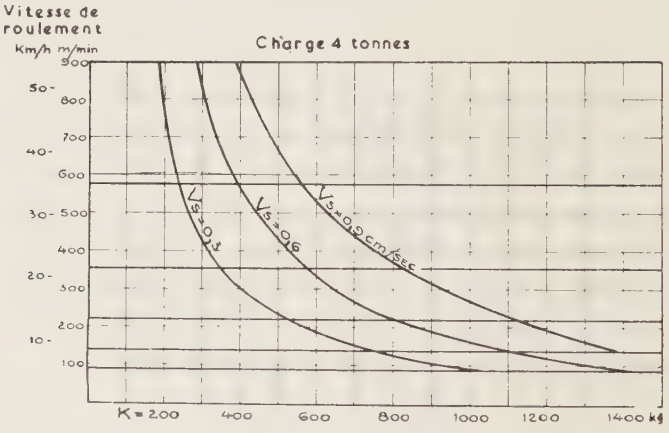


Vs = Speed of transverse displacement.
K = Force of transverse displacement.

Fig. 22.

Explanation of French terms :

Vitesse de roulement = Running speed. — Charge 2 tonnes. = Load 2 metric tons.



Vs = Speed of transverse displacement
K = Force of transverse displacement.

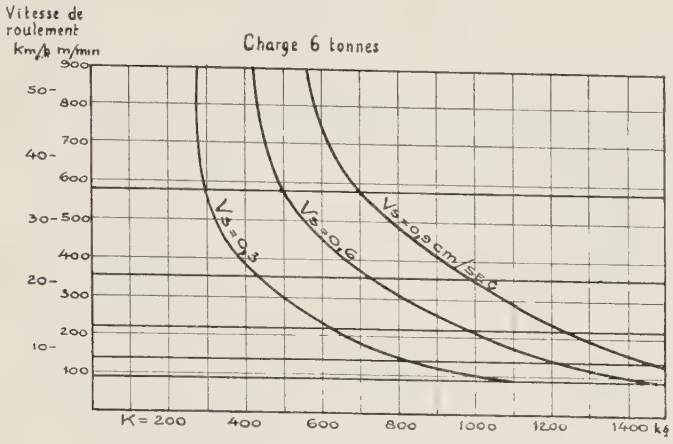
Fig. 23.

Explanation of French terms :

Vitesse de roulement = Running speed. — Charge 4 tonnes = Loading 4 metric tons.

transmitted by an angle lever K with a pivot M.

The value of the force exercised by this lever was recorded graphically on the table N, fixed to the yoke C, by displacement in a manner homothetic to the upper wheels. The pen, directed in opposite direction, vertically in the frame. was



Vs = Speed of transverse displacement.
K = Force of transverse displacement.

Fig. 24.

Explanation of French terms :

Vitesse de roulement = Running speed. — Charge 6 tonnes = Load 6 metric tons.

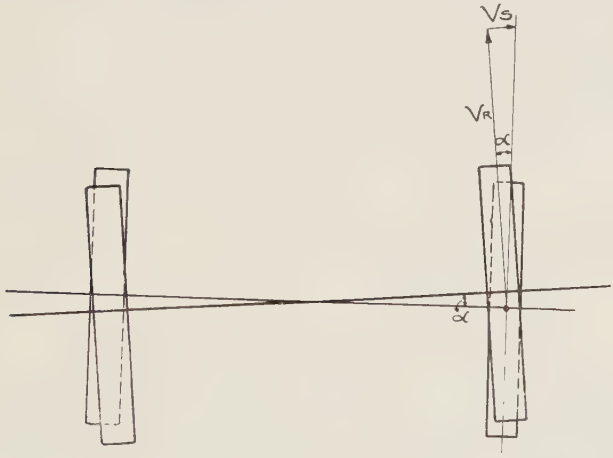


Fig. 25.

coupled to the spring G, so that the vertical displacement was proportionate to the tension of the dynamometer. The transverse movement of the upper set of wheels was limited by the stops S, fixed to the frame.

Later, the installation was modified so that the dynamometer was tensioned by mechanical drive from an axle turning at a constant speed, with various transmission ratios and thus at various speeds which could be determined exactly. The mechanical drive of the dynamometer was necessary to make possible recording

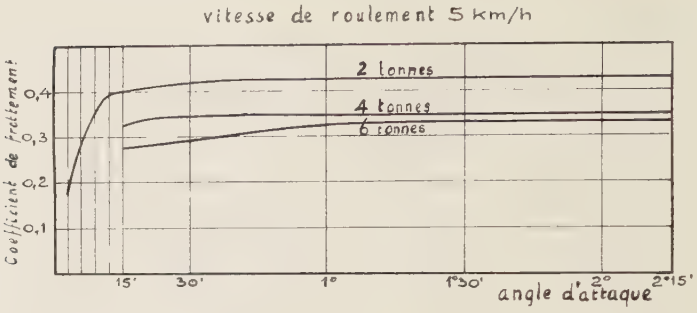


Fig. 26.

Explanation of French terms :

Coefficient de frottement = Coefficient of friction. — Vitesse de roulement 5 km/h = Speed of movement 5 km/h. — Angle d'attaque = Angle of bearing.

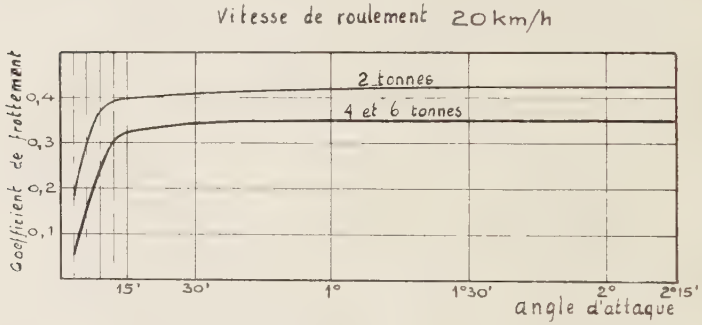


Fig. 27.

Explanation of French terms :

Coefficient de frottement = Coefficient of friction. — Vitesse de roulement 20 km/h = Speed of movement 20 km/h — Angle d'attaque = Angle of bearing.

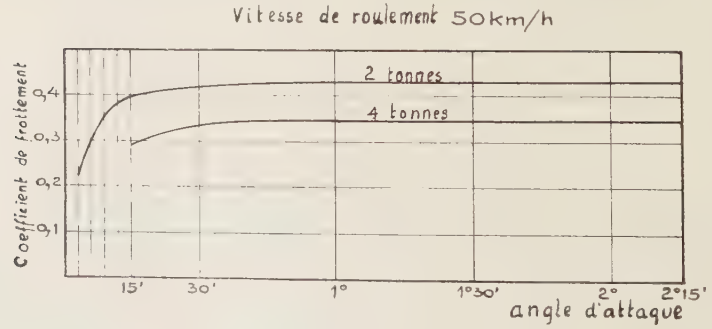


Fig. 28.

Explanation of French terms :

Coefficient de frottement = Coefficient of friction. — Vitesse de roulement 50 km/h = Speed of movement 50 km/h. — Angle d'attaque = Angle of bearing.

of the speed of transverse slide in relation to the speed of rotation. Figure 20 shows the modified construction diagrammatically.

Figure 21 is a specimen diagram from which can be deduced the sliding force and speed. A is the initial point.

The upper pair of wheels bear on the angle lever K (fig. 20) and cover the distance AB without exercising any force whatever. At the point B, the sliding force comes into action; at first the axle is not moved. At the point C displacement of the axle commences under pressure from the dynamometer which increases continuously up to a position corresponding to D where the maximum value of the action of the force is reached. Beyond this, displacement of the axle takes place with a constant force. The speed of slide can be determined by means of the points shown on the diagram, which are marked at 0.5 second intervals.

In this way, a certain number of tests were made with axle loads of 2.4 and 6 tons at various speeds of rotation and slide. The results obtained are shown graphically in figures 22-24.

It is concluded from this that the coefficient of friction f' for constant speeds of rotation and slide decreases as the load increases. For example with a running speed of 400 m/min (24 km/h) and a sliding speed of 0.9 cm/sec (5.4 m/min), the values are :

with a load of 2 tons : $f' = 0.250$;

with a load of 4 tons : $f' = 0.187$;

with a load of 6 tons : $f' = 0.154$.

The rotary movement of the wheels was correct to 0.1 mm, whilst the parallel mounting of the two sets of wheels excluded the possibility of any obliqueness whatsoever.

To overcome the forces of inertia, trials were pursued, the upper set of wheels being mounted at a certain angle to the lower wheels. The trials were carried out with angles α of 3′ to 2°15′.

Assuming that the upper set of wheels moves at a speed V, its sliding speed should be $Vs = V_R \operatorname{tg} \alpha$, so it remains in effect in the same place (see fig. 25).

Here we have reproduced exactly the case of a wheel bearing on the rail. The trials were carried out at running speeds of 5, 20 and 50 km (3, 12 and 31 miles)/h. The results are shown in Table IV and are graphed in figures 26 to 28.

These illustrations allow us to conclude that the friction to transverse slide of a rotating wheel, or what comes to the same thing, when the rotation of a wheel running at a certain angle on the rail, can reach a high enough value if the speed of slide is relatively high. Friction was so pronounced that the surfaces gripped and small particles were detached, coming away in their original polished condition.

These two sets of tests, one with parallel pairs of wheels and the other with wheels set at relative angles, gave similar results as regards the coefficient of friction. It also showed that the absolute value of the coefficient of friction f' for a given axle load, depends solely on the relationship between the speeds of rotation and slide. This relationship is shown in figure 29.

The results of the trials carried out may be summarised as follows :

If two cylindrical rollers with parallel axes run one on the other at a speed of

TABLE IV.

	Angle of bearing	Load 2 metric tons		Load 4 metric tons		Load 6 metric tons	
		K	f'	K	f'	K	f'
		kg	—	kg	—	kg	—
Speed of movement 5 km/h	3′	348	0.174	—	—	—	—
	6′	564	0.282	—	—	—	—
	9′	720	0.360	—	—	—	—
	12′	792	0.396	—	—	—	—
	15′	798	0.399	1 290	0.324	1 640	0.274
	30′	840	0.420	1 374	0.345	1 730	0.288
	1°	852	0.426	1 380	0.346	1 962	0.327
	1° 30′	858	0.429	1 380	0.346	1 980	0.330
	2°	858	0.429	1 380	0.346	1 980	0.330
	2° 15′	858	0.429	1 380	0.346	1 980	0.330
Speed of movement 20 km/h	3′	378	0.189	216	0.054	—	—
	6′	594	0.297	648	0.162	—	—
	9′	745	0.373	960	0.240	—	—
	12′	786	0.393	1 248	0.312	—	—
	15′	798	0.399	1 314	0.329	1 920	0.320
	30′	822	0.411	1 398	0.349	2 130	0.355
	1°	840	0.420	1 416	0.354	2 130	0.355
	1° 30′	852	0.426	1 416	0.354	2 130	0.355
	2°	852	0.426	1 416	0.354	2 130	0.355
	2° 15′	852	0.426	1 416	0.354	2 130	0.355
Speed of movement 50 km/h	3′	444	0.222	—	—	—	—
	6′	600	0.300	—	—	—	—
	9′	708	0.354	—	—	—	—
	12′	768	0.384	—	—	—	—
	15′	792	0.396	1 170	0.292	—	—
	30′	840	0.420	1 338	0.335	—	—
	1°	864	0.432	1 380	0.345	—	—
	1° 30′	864	0.432	1 380	0.345	—	—
	2°	864	0.432	1 380	0.345	—	—
	2° 15′	864	0.432	1 380	0.345	—	—

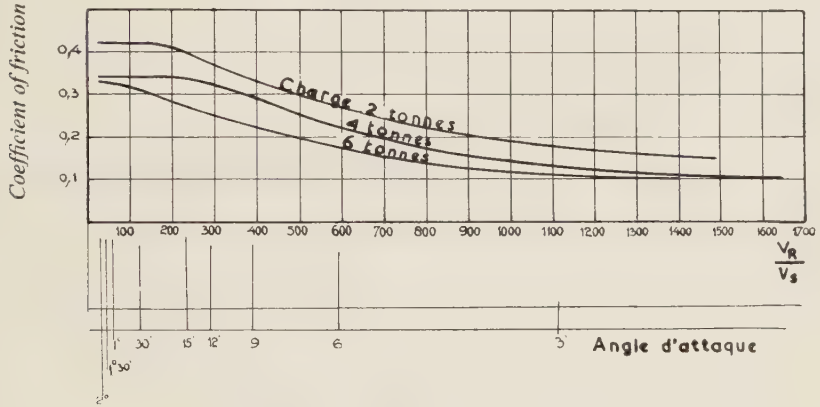
V_R and if one of the cylinders slides at the same time on the other in the direction of their axis at a speed of slide Vs, or, what is really the same thing, if one cylindrical roller runs on another at the same running speed V_R whilst their axes form a certain angle, the coefficient of friction depends in this case solely on the relationship V_R : Vs with the surface pressure remaining constant and the materials being identical.

The latter result is very important, since

it will allow the following trials to be confined to axles in an oblique position and a restrained running speed as $Vs = V_R \operatorname{tg} \alpha$; the absolute value of V_R is not an important factor.

The trials were subsequently continued with axle loads of 8 and 10 tons, the the treads of the lower set of wheels being rounded off with a 400 mm (1′ 3 3/4″) actual conditions, a three-axled standard gauge vehicle was built to enable trials to continue (fig. 34).

One of the outer axles A has no transverse play and is therefore a so-called rigid axle. This axle carries a normal load. The other outer axle is also rigid but has no suspension springs as it is not loaded and serves only to guide the



$$\frac{V_R}{Vs} = \text{ratio } \frac{\text{speed of movement}}{\text{speed of transverse displacement}}$$

Fig. 29.

Explanation of French terms :
Charge 2 tonnes = Load 2 metric tons. — Angle d'attaque = Angle of bearing.

radius, similar to that of the rail head in use (see fig. 30 and 31).

Figures 32 and 33 show the contact areas in the case of a coned wheel on a convex surface of 400 mm radius and the same wheel on a rail rounded off at 400 mm.

Test vehicle for determining the coefficient of friction between wheel and rail.

As we have noted, the early trials did not reproduce actual conditions, since a circle was running on a circle and not on a straight line. To approach nearer to vehicle in the track. The axles A and B are flanged to give a minimum of play in the track.

The intermediate axle C comprises a cylindrical tread without flange. The load on this axle is transmitted by means of mountings D on the axlebox and in this way to the wheels and to the rails. The axle has lateral displacement and can also take up an oblique position. This is the measuring axle or as it is otherwise called, the test axle. Transverse movement is limited by a lever E

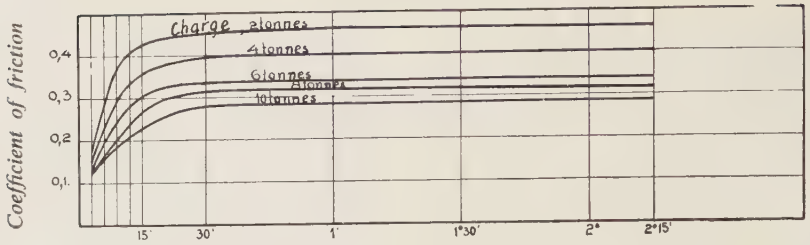
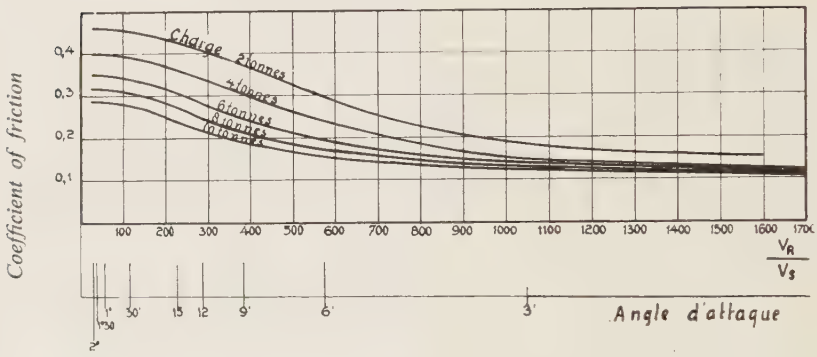


Fig. 30.



$$\frac{V_R}{V_S} = \text{ratio } \frac{\text{speed of movement}}{\text{speed of transverse displacement}}$$

Fig. 31.

Explanation of French terms :

Charges 2 tonnes = Load 2 metric tons. — Angle d'attaque = Angle of bearing.

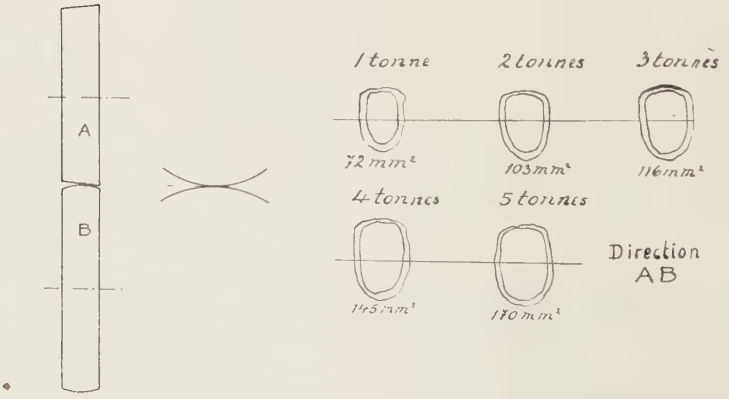


Fig. 32.

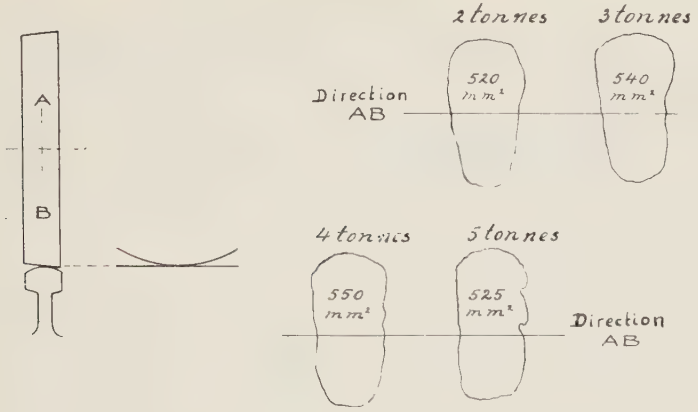


Fig. 33. — Determination of areas of contact.

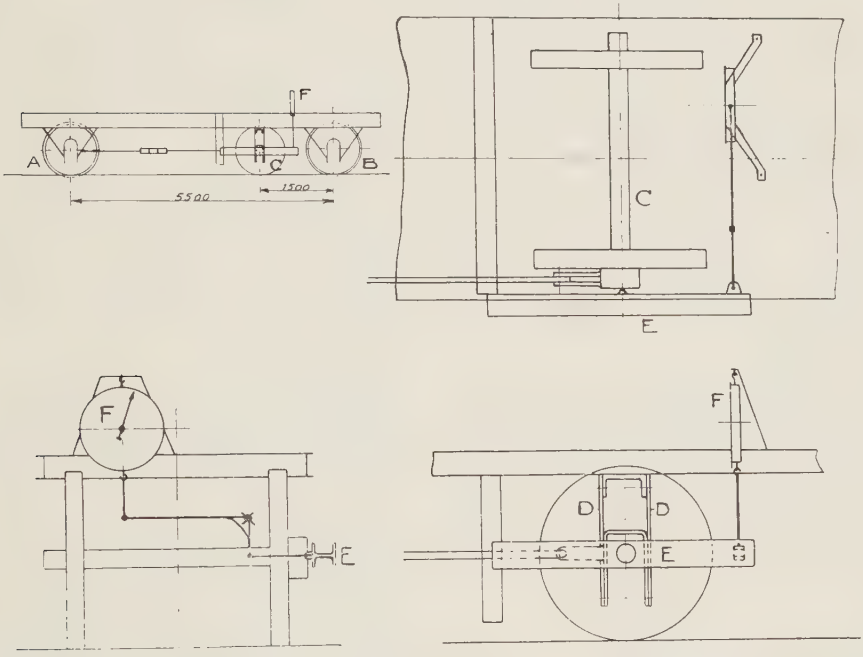


Fig. 34.

which exercices thrust along the axle centre line and is loaded by a dynamometer F.

If the vehicle runs forward with a position oblique to the axle C, the axle will tend to overrun the rails since the vehicle is guided in the track by axles A and B.

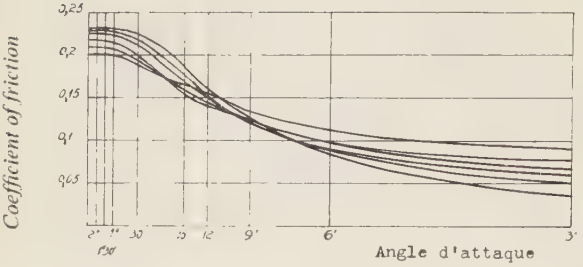


Fig. 35. — Two wheels of equal diameters.

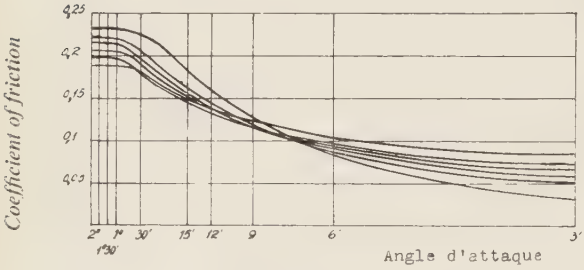


Fig. 36. — Difference between the 2 wheel diameters 1 mm.

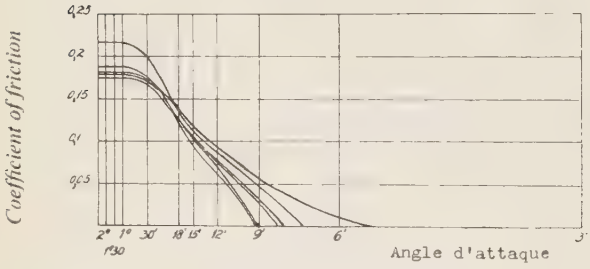


Fig. 37. — Difference between the 2 wheel diameters 3 mm.

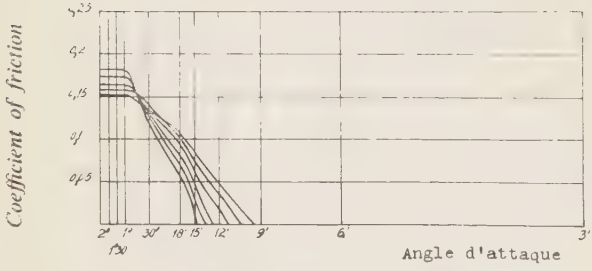


Fig. 38. — Difference between the 2 wheel diameters 5 mm.

Angle d'attaque = Angle of bearing.

The forces working between the wheels of axle C and the rails are transmitted by the axle and the lever E to the dynamometer. This transverse force can be determined exactly. To provide for obtaining the dynamic load, axle B is not designed to support any part of the weight of the vehicle.

In order to reproduce the action of tyres with conical tread section, the above tests were continued with axle C having cylindrical tread sections, the diameter of one of the wheels being less by 1, 3 or 5 mm (3/64″, 1/8″ or 13/64″) than the other wheel. One of the wheels was thus running a different circle to the other, which was equivalent to a set of wheels displaced transversely with conical tread.

As regards the results obtained with cylindrical wheels of equal diameter, compared with those on the test bench, values of f' in the upper oblique position were somewhat less with the test vehicle than on the bench.

The test of the wheels of axle C, with different diameters showed that conical treads had a good influence. Values of f' for conical tyres were less than those with cylindrical tyres. The value of f' decreases as conicity increases in such a way that with a 1 : 20 coning (difference in wheel diameters, 5 mm) at an oblique position from 0 to 12′ at which the value of $f' = 0$, the value only increased to 0.15 for an oblique position of 2°15′.

The results obtained with wheels of unequal diameters should be considered as fictitious results, so far as the value of f' is concerned, since we do not know

TABLE V.

Angle of bearing of axle C	Load on axle C											
	6 t		8 t		10 t		12 t		14 t		16 t	
	K	f'	K	f'	K	f'	K	f'	K	f'	K	f'
	kg	—	kg	—	kg	—	kg	—	kg	—	kg	—
3′	200	0.03	400	0.05	600	0.06	850	0.07	1 100	0.08	1 450	0.09
6′	500	0.08	700	0.09	900	0.09	1 150	0.10	1 350	0.09	1 800	0.11
9′	750	0.13	950	0.12	1 200	0.12	1 500	0.13	1 750	0.13	2 100	0.13
12′	950	0.16	1 200	0.15	1 450	0.15	1 650	0.14	2 000	0.14	2 500	0.16
15′	1 120	0.19	1 400	0.18	1 700	0.17	1 850	0.16	2 200	0.16	2 650	0.17
30′	1 350	0.23	1 700	0.22	2 100	0.21	2 400	0.20	2 700	0.19	3 000	0.19
1°	1 400	0.23	1 850	0.23	2 250	0.23	2 600	0.22	2 900	0.22	3 250	0.20
1° 30′	1 400	0.23	1 850	0.23	2 250	0.23	2 600	0.22	2 900	0.22	3 250	0.20
2°	1 400	0.23	1 850	0.23	2 250	0.23	2 600	0.22	2 900	0.22	3 250	0.20
2° 15′	1 400	0.23	1 850	0.23	2 250	0.23	2 600	0.22	2 900	0.22	3 250	0.20

to what extent the transverse distance to be covered in a sliding movement changes by reason of the conical shape of the tyres and to what extent this shape reduces the grinding force.

The results of these tests are shown in Tables V to VIII, and graphically in figures 35-38.

We can conclude from them that with an oblique position of more than 1° the coefficient of friction f' does not vary. Table IX gives the values of f' for the different cases.

The action of the forces is reproduced in figures 39 to 41.

If the two wheels of the axle C have the same diameter, this axle will be found to have an oblique position as in figure 39, and the vehicle will run in the direction of the arrow N; the wheels of axles A and B will exert on the left-hand rail the forces K_A and K_B :

$$K_A = \frac{b}{a+b} K$$

and :

$$K_B = \frac{a}{a+b} K.$$

The two wheels of axle C run in a

TABLE VI.

Angle of bearing on axle C	Load on axle C											
	6 t		8 t		10 t		12 t		14 t		16 t	
	K	f'	K	f'	K	f'	K	f'	K	f'	K	f'
	kg	—	kg	—	kg	—	kg	—	kg	—	kg	—
3′	200	0.03	400	0.05	600	0.06	800	0.07	1 000	0.07	1 350	0.08
6′	500	0.08	700	0.09	900	0.09	1 100	0.09	1 300	0.09	1 700	0.11
9′	750	0.13	950	0.12	1 150	0.12	1 400	0.12	1 600	0.12	2 000	0.13
12′	950	0.16	1 150	0.14	1 350	0.14	1 600	0.14	1 900	0.14	2 300	0.14
15′	1 100	0.18	1 300	0.16	1 600	0.16	1 800	0.15	2 100	0.15	2 500	0.16
30′	1 350	0.23	1 650	0.21	1 950	0.20	2 300	0.19	2 550	0.18	2 900	0.18
1°	1 400	0.23	1 750	0.22	2 150	0.22	2 500	0.21	2 750	0.20	3 000	0.19
1° 30′	1 400	0.23	1 750	0.22	2 150	0.22	2 500	0.21	2 750	0.20	3 000	0.19
2°	1 400	0.23	1 750	0.22	2 150	0.22	2 500	0.21	2 750	0.20	3 000	0.19
2° 15′	1 400	0.23	1 750	0.22	2 150	0.22	2 500	0.21	2 750	0.20	3 000	0.19

direction longitudinal to the track and slide in a transverse direction.

On the other hand, if the two wheels of axle C have unequal diameters, and if this axle does not have an oblique position as in figure 40, the smaller diameter wheel will slide and exert a force T on the right-hand rail, more or less in the longitudinal direction of the track. The wheel of larger diameter will supply the reaction to this force exerted by the wheel of smaller diameter, so that a force T also acts on the left-hand rail but in an inverse direction.

The two forces T form a moment :

$$M = \ddot{c}\ T$$

The forces shown in figure 40 are exerted by the vehicle in the track. On the vehicle, therefore, the moment operates inversely, consequent upon the action of this moment, the axle A is thrust with a force K'_A against the right-hand rail and the axle B with a force K'_B against the left-hand rail.

The forces have the following values :

$$K'_A = \frac{C}{a + b}\ T$$

TABLE VII.

Angle of bearing on axle C	Load on axle C											
	6 t		8 t		10 t		12 t		14 t		16 t	
	K	f'	K	f'	K	f'	K	f'	K	f'	K	f'
	kg	—	kg	—	kg	—	kg	—	kg	—	kg	—
3′	—	—	—	—	—	—	—	—	—	—	—	—
6′	—	—	—	—	—	—	—	—	—	—	240	0.02
9′	—	—	—	—	250	0.03	350	0.03	650	0.05	880	0.06
12′	400	0.07	500	0.06	700	0.07	900	0.08	1 200	0.09	1 500	0.09
15′	600	0.10	750	0.09	1 000	0.10	1 200	0.10	1 500	0.11	1 850	0.12
18′	800	0.13	1 000	0.13	1 250	0.13	1 500	0.13	1 800	0.13	2 200	0.14
30′	1 200	0.20	1 400	0.18	1 700	0.17	2 100	0.18	2 400	0.17	2 700	0.17
1°	1 300	0.22	1 500	0.19	1 800	0.18	2 200	0.18	2 500	0.18	2 800	0.18
1° 30′	1 300	0.22	1 500	0.19	1 800	0.18	2 200	0.18	2 500	0.18	2 800	0.18
2°	1 300	0.22	1 500	0.19	1 800	0.18	2 200	0.18	2 500	0.18	2 800	0.18
2° 15′	1 300	0.22	1 500	0.19	1 800	0.18	2 200	0.18	2 500	0.18	2 800	0.18

and :

$$K'_B = \frac{C}{a+b} T$$

If the axle C takes an oblique position, the wheels not having the same diameters (see fig. 41), one wheel of the axle A then thrusts against the left hand rail with a force :

$$K''_A = K'_A - K_A$$

and one wheel of axle B against the left-hand rail with a force :

$$K''_B = K_B + K'_B$$

The force K''_A can be nil, or negative, i. e. the pressure is exercised by the other wheel of A on the right hand rail.

It has been seen with the test vehicle that with an oblique position of about 50′ the force K''_A becomes nil. If the oblique position is less than 50′, it is the left-hand wheel which bears on the rail and if the oblique position exceeds this figure, the right-hand wheel which bears on the rail.

What is the explanation of force K″ being less than K ?

TABLE VIII.

Angle of bearing on axle C	Load on axle C											
	6 t		8 t		10 t		12 t		14 t		16 t	
	K	f'	K	f'	K	f'	K	f'	K	f'	K	f'
	kg	—	kg	—	kg	—	kg	—	kg	—	kg	—
3′	—	—	—	—	—	—	—	—	—	—	—	—
6′	—	—	—	—	—	—	—	—	—	—	—	—
9′	—	—	—	—	—	—	—	—	—	—	—	—
12′	—	—	—	—	—	—	200	0.02	500	0.04	800	0.05
15′	—	—	200	0.03	400	0.04	700	0.06	1 000	0.07	1 300	0.08
18′	350	0.06	550	0.07	800	0.08	1 100	0.09	1 400	0.10	1 700	0.11
30′	700	0.12	1 000	0.13	1 300	0.13	1 600	0.14	1 800	0.13	2 100	0.13
1°	1 100	0.18	1 400	0.18	1 650	0.17	1 900	0.16	2 150	0.16	2 500	0.16
1° 30′	1 100	0.18	1 400	0.18	1 650	0.17	1 900	0.16	2 150	0.16	2 500	0.16
2°	1 100	0.18	1 400	0.18	1 650	0.17	1 900	0.16	2 150	0.16	2 500	0.16
2° 15′	1 100	0.18	1 400	0.18	1 650	0.17	1 900	0.16	2 150	0.16	2 500	0.16

TABLE IX.

Difference in diameters of wheels on axle C	Position 1° oblique with axle load in metric tons					
	6	8	10	12	14	16
	valeur de f'					
0 mm	0.23	0.23	0.23	0.22	0.22	0.20
1 mm	0.23	0.22	0.22	0.21	0.20	0.19
3 mm	0.22	0.19	0.18	0.18	0.18	0.18
5 mm	0.18	0.18	0.17	0.16	0.16	0.16

In the case of figure 39 the two wheels of axle C run in the direction of the track and slide transversely; force K causes the transverse slide. In the case of figure 41, the larger diameter wheels runs on the rail, due to the unequal diameters. We have here two directions of slide. one across the other. The force K″ as measured determines only the force of friction in the transverse direction and this, for the

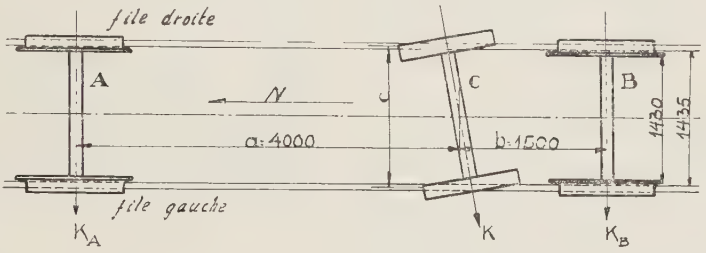


Fig. 39.

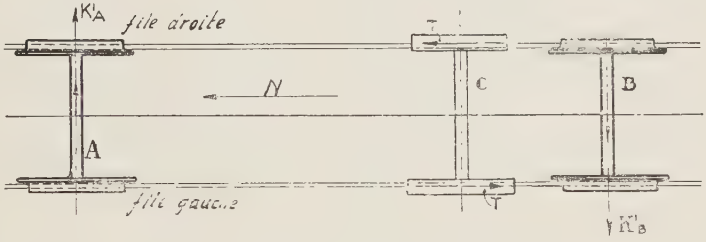


Fig. 40.

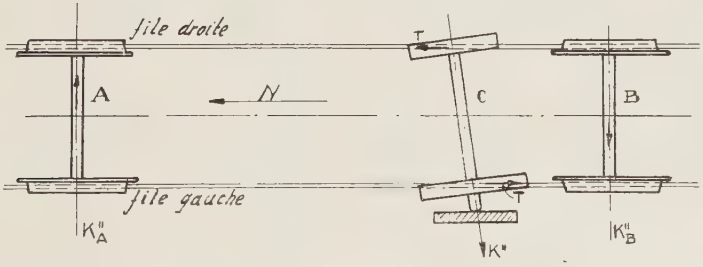


Fig. 41.

Explanation of French terms :
File droite — Right hand rail — File gauche — Left hand rail.

left-hand wheel and slides across it; the right hand wheel slides longitudinally to the rail and also transversely; the sliding accross is done more easily because of the longitudinally slipping along the smaller diameter wheel, is weaker than in the case of figure 39, and, for the larger diameter wheel, equal to that of figure 39.

(To be continued.)

[625 .143 .4]

Slightly staggered rail joints,

by A. Saldanha,

M. A. Cantab., M. I. E. India, Fellow P. W. Inst., Chief Engineer, G. I. P. Railway.

FOREWORD.

The logic of staggering rail joints is irresistible. The joint is recognised as a weak part of the track structure and it is merely sound horse-sense to avoid placing two weak spots directly opposite each other. But till recently, staggering of joints has been confined only to curves, and there too the joints are laid to a « mid-stagger » *i.e.* half a rail length apart.

This study is a reprint of two articles contributed by the author to the *Quarterly Technical Bulletin* published by the Railway Board. It sets out the considerations in favour of adopting a « short » stagger on all track, both straight and curved, explains how the objections put forward were overcome by the invention of a bifurcated tie-bar, and describes the development of the installation of slightly staggered rail joints on the G. I. P. Railway. Finally, it goes on to derive the optimum « pitch » of the stagger by a process of analytical deduction.

The purpose of this article is to enable engineers and inspectors to follow clearly the reasons why the use of the short stagger is advocated. It is hoped it will also convince any « doubting Thomases » among track specialists of their real merits and decided superiority over square joints.

In addition to these merits, staggering of joints also has the virtue of being a useful anti-sabotage device, as unlike track laid with square joints, it prevents the track being broken and slued even when the fishplates are removed. The bifurcated tie-bar further assists in maintaining the continuity of the track structure, and making the removal of the fishplates innocuous. In these days where attempts at sabotage are not uncommon and saboteurs resort to the simple expedient of removing the fishplates, any device which makes the removal of fishplates less hazardous for the travelling public merits careful consideration from the point of view of introducing a safety complex in track design.

Slightly staggered rail joints.

The staggered joints with which track engineers are familiar is the « half-stagger » used on curves, partly to overcome the difficulty of obtaining square joints, because of the lead of the inner rail, and partly to secure a smoother alignment by preventing elbowing at the joints, noticeable with F. F. rails of heavy section, which do not naturally lend themselves to lateral flexure.

The effect of staggering is also to reduce the impact at joints to that corresponding to a wheel load, instead of an axle load, and to produce more uniform vertical continuity of the track.

Prima facie, therefore, it can be concluded that the staggering of rail joints is beneficial in providing better lateral and vertical continuity of the track, both on curves and on the straight.

Due partly to weak fishplates, and partly to inadequate support with the comparatively wide sleeper spacing at the joints, the joints of obsolescent track are mostly low, and the occurrence of these low spots at regular intervals of half a rail length on alternate rails has been found to set up an undesirable and uncomfortable rolling motion in locomotives and rolling stock, owing to the regular periodicity of the roll on either side, particularly when it synchro-

nises with the period of vibration of the springs.

The problem, therefore, is to obtain the benefits of staggering the joints, while at the same time eliminating the hazard and discomfort of synchronous rolling motion.

At Meeting N° 11 (1940) of the Engineering Section of the I. R. C. A., the Chief Engineer, North Western Railway, who was appointed Reporter on subject N° N/264 « Staggered rail joints on straight track », described a method of achieving this by staggering the rail joints a distance less than the dimension between the closest spaced bogie wheels running on the Railway.

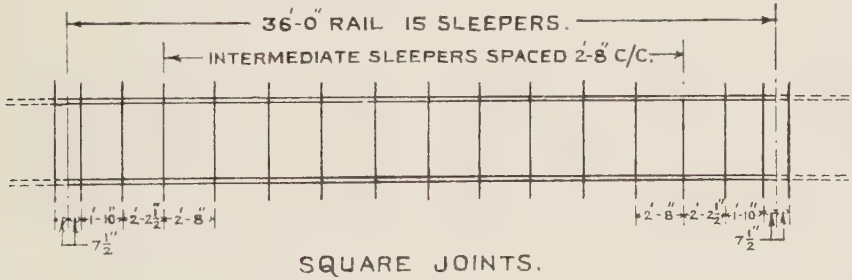
« The effect of such a stagger is that the four wheels of a single bogie will pass over the joints with an alternate right and left-hand vertical displacement on front and rear wheels. Assuming the joint in the right hand rail is leading, first the left leading wheel, then the right leading, followed by the left trailing and finally the right trailing wheel, will pass over the joints in a manner which can best be referred to as a « wriggling » motion. This transition over the joints may be compared with the motion of a motor car passing over a ridge in a road diagonally instead of on the straight. It is well known that by negotiating an obstruction in such a manner the uncomfortable effects on the passengers in the motor car are very considerably reduced . . . . . .

. . . . . .

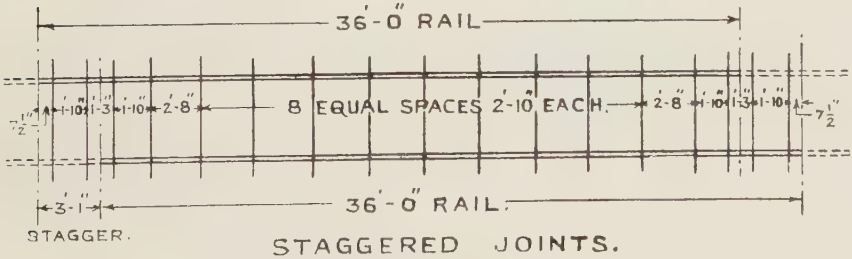
« The beneficial effect on running has been very noticeable and the joints are negotiated with the minimum of impact or vertical movement transmitted to the coach itself. » (1).

The sleeper density adopted in the experiment was (N+3) standard for 36 ft. rails, and a comparison of the sleeper spacing of track laid with square joints, and track laid with slightly staggered (3'-1") joints is shown below, for C. I. sleepers :

(1) Page 37 of Report of the I. R. C. A. Engineering Section, Meeting N° 11.

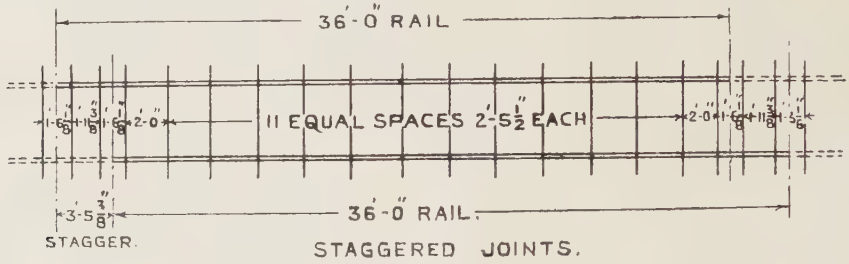


SQUARE JOINTS.



STAGGERED JOINTS.

The G. I. P. Engineers were quick to recognise that the short stagger had great possibilities, and adopted sligthly staggered joints as the standard on relaying, with a sleeper density of (N+4) for 36 ft. rails to the spacing shown below :



STAGGERED JOINTS.

At a subsequent meeting (1), however, the Chief Engineer, N. W. R., submitted the following adverse report :

« Records of running conditions on lengths of track laid with staggered joints were taken during the year with the Hallade instrument. The results obtained do not show any advantage in favour of staggered joints; in fact, square joints show slightly better results. Previous impressions of good running on staggered joints may have been due to the new sleepers then laid in the track and partly because the running is less noisy, as both wheels do not strike the joints at the same time.

» The staggering of rail joints involves a bunching of the sleepers under the joints and, consequently, wider spacing of sleepers in the centres of the rails, as compared with spacings with square joints. Such bunching and non-uniform spacing of sleepers means a varying elastic modulus of the track. Moreover, at the joints the spacing of the sleepers between the joints leads to greater support of the rail on one side of a joint as compared with the support on the other side. On account of the bunching of the sleepers at the joints and the wider spacing of the sleepers in the centres of the rails, there is constant trouble from low centres of the rails. This leads to uncomfortable running in addition to the rolling motion of the joints. It is felt that this will get worse as the sleepers become older.

» It is not intended to lay any more track with staggered joints and, as this type of track does not give any better running compared with square joints, it is not proposed to submit any further reports on this experiment. » (1).

With the hazard of synchronous rolling eliminated by the substitution of the short stagger for the half-stagger, the beneficial effects of staggering (2) are so decided, that the problem of formulating a solution which will retain the benefits of the short stagger and at the same time overcome the valid objections put forward by the N. W. R., is worth serious study. The problem in a nut shell with slightly staggered joints is to reproduce the accepted efficient rail joint support for square joints but without any bunching of sleepers under the continuous rail opposite the joint, and without too wide a spacing of the inter-

(1) Meeting N° 13 (1943) of the Engineering Section of the I. R. C. A.

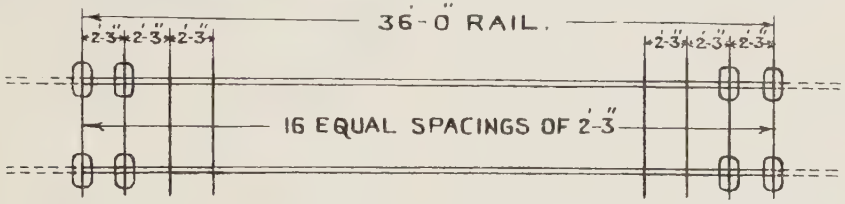
(1) Report dated 21st October, 1942, by A. M. SIMS, Chief Engineer, N. W. Rly. at the 13th meeting of the Engineering Section of the I. R. C. A., 1943.

(2) « In America, there exists the wise rule of setting the joints out alternately instead of opposite one another. In Europe such a disposition of the rail joints is only now beginning to be used ». Introduction to Railway Mechanics : George V. LOMONOSSOFF.

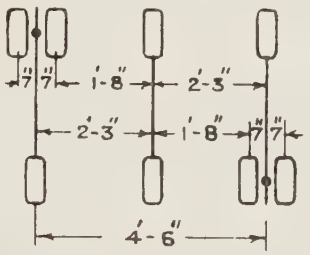
mediate sleepers, *i.e.*, a uniformly elastic track structure-cumslightly staggered joints.

Before describing the proposed solution, it is necessary to specify what is the most efficient rail joint support using standard fishplates and sleepers and without resort to expensive patent devices. According to the world consensus of opinion and experience (1), the most efficient rail joint support is secured by laying the joint sleepers practically touching, the shoulder sleeper at about 1'-9", the next to shoulder sleeper at 2'-0" to 2'-3", the spacing of the intermediate sleepering depending on the density of the sleepering.

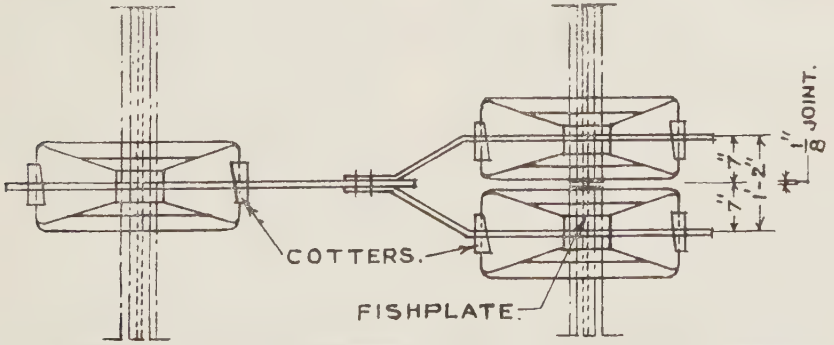
Consider now a 36 ft. rail length supported on 17 C. I. plate sleepers, equally spaced at 2'-3" centres :



Sub-divide the first and second spacing of 2'-3" into two parts, measuring 7" and 1'-8" respectively in the manner shown diagrammatically below :

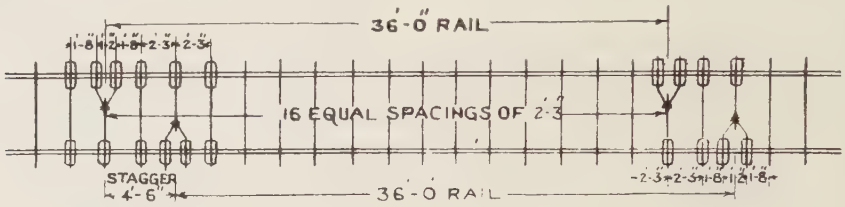


All that is required to translate the above into practice with C. I. plates, is to use a bifurcated tie-bar at the joints, with two C. I. plates threaded at the end supporting the joint, and one C. I. plate at the opposite end under the continuous rail :

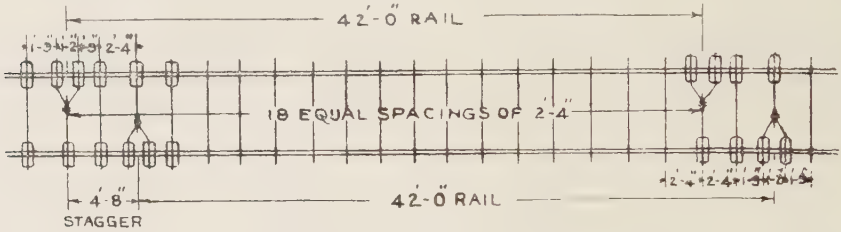


(1) See proceedings of the Paris Congress, 1937, *Bulletins of the International Railway Congress;* also subject N° N/267 « The support of rail joints » in the proceedings of the I. R. C. A., Engineering Section.

The sleeper spacing with a density of (N + 5) for 36 ft. rails, and the support in the vicinity of the joints, with a short stagger of 4′-6″ is shown below for C. I. plates of the CST 9 type.



Similarly for 42 ft. rails, the same symmetrical spacing can be derived by dividing the distance into 18 equal spacings of 2′-4″.



Notice that each joint is supported by two closely spaced plates, with the shoulder sleeper on either side a about 1′-9″. The bifurcated tie-bar permits the use of two plates at the rail joint, and of one plate at the point opposite the joint on the continuous rail, thereby overcoming the objection of the hard spot on the rail opposite the joint.

One has only to compare the sleeper spacings used in the preliminary trials on the N. W. R. and G. I. P. and reproduced earlier in this article, with the spacings now proposed, to be convinced of the improvement possible in securing a symmetrical spacing of sleepers and a uniformly elastic modulus of track support by the simple expedient of using at the rail joints bifurcated tie-bars, which can be turned out in any small workshop. Exactly the same number of sleepers are used as would be required to secure the same modulus of track support with square joints. At the same time, this arrangement also indicates the optimum spacing of intermediate sleepers to secure a uniformly elastic track structure. For, if the closest intermediate spacing possible on the continuous rail opposite the rail joint is 2′-3″, there is no object in reducing this distance further along the rail by increasing the density of the sleepering, as it would only result in a track of varying longitudinal modulus of elasticity, hard in the intermediate portions, and yielding in the zone of the joints.

As regards the amount of the stagger, the minimum rigid wheel base permitted by Standard Dimensions for bogie trucks of any vehicle on the broad gauge is 6 ft., so that the condition that the amount of the stagger should be less than the dimension between the closest spaced bogie wheels is satisfied. As a matter of fact, the normal wheel base for bogie trucks of broad gauge passenger rolling stock on Indian Railways is 10 ft.

On curves, adopting the usual practice of using a cut rail, short by the pitch of the fishbolt holes (4 ½″ for 90 lbs. R. F. F. track) when the lead of the inner rail

becomes half the pitch of the bolt holes (2 ¼″ for 90 lbs. R. F. F. track), the extent of the stagger will vary from that on the straight by ± half the pitch of fishbolt holes. The sleeper spacing between the staggered joints will require slight adjustment to distribute this variation.

This drawing-board solution is now being given a practical trial under fast operating conditions on the G. I. P. on a 50 mile stretch of relaying. The test will indicate to what extent the arrangement stands up in practice as regards alignment, gauge, level and joint support. This cannot be predicted on the drawing board.

The arrangement has the advantage of simplicity. A symmetrical and uniformly elastic support is secured without the use of special types of joint sleepers, fishplates and fastenings. The only non-standard fastening employed is the bifurcated tie-bar, which is too simple to manufacture, if necessary from discarded full length tie-bars. It may, therefore, be of interest to other railways who are using or experimenting with slightly staggered joints. The arrangement is applicable only to track laid with C. I. plates connected by means of a tie-bar.

Criticism will probably be directed to the efficacy of the bifurcated tie-bar as a gauge link, also its durability under torsional stresses. Here again, it may be stated that although there is no difficulty in providing a distance piece at the end of the bifurcation, it has been deliberately left out in the first instance, as it may turn out to be unnecessary in actual practice, once the plates take a stable bedding.

1. What is the optimum stagger?

In an article which appeared in the January 1947 issue of the Quarterly Technical Bulletin, the author discussed the claims made for the « Short-stagger » and the design of a composite cast-iron sleeper with a bifurcated tie-bar to secure uniform bearing pressure on the sub-formation at joints and along the rail when the joints are staggered.

2. The idea of laying the track to a « short-stagger » has, since the publication of this article, aroused worldwide interest, and the author has received numerous enquiries both from home and foreign railways regarding the merits and the practical features of track laid with slightly-staggered joints. He therefore proposes to give a resume of the considerations leading to the adoption of the short-stagger and up-to-date developments of designs of composite sleepers for use with track laid to a short-stagger.

3. Going back to history, he would mention that on the G. I. P. Ry., it had been accepted that the short-stagger was good, both on straight and curved track. The first problem he set himself to solve was to secure a uniform modulus of track support with staggered joints. It involved the design of a composite joint sleeper which, while providing adequate joint support at the rail joint end, provided only the normal support under the continuous rail at the opposite end. In his previous article he described the design of this composite sleeper.

4. The next problem which exercised his mind was this: The dimension of short-stagger used was 3′-5 3/8″. What was the virtue of this dimension? Assuming that it is good to stagger rail joints but that the mid-stagger is undesirable, what is the optimum stagger considering both the track and the rolling stock as a whole? He will now describe how he arrived at the optimum dimension for the short-stagger by a process of logical deduction and without the use of formidable equations and calculas symbols which are apt to baffle practical technicians.

5. Before dealing with the question of the optimum stagger of rail joints, the author will briefly recapitulate the arguments in favour of the short-stagger.

The advantage of staggering joints (to any pitch) is that it produces better lateral as well as *vertical continuity* of the track. The improved lateral continuity is evi-

denced on curves where it will be observed that staggering of joints produces a smooth curve, whereas square joints have a tendency to produce elbowed track. The improved vertical continuity can be observed by suspending a miniature model track, first with square joints and next with staggered joints, when it will be seen that the track set with staggered joints, hangs in a smoother and a flatter vertical curve than track with square joints.

6. The effect of staggering (to any pitch) is also to reduce the impact on the rolling stock and track structure at a given instant considerably, as compared to track laid with square joints. The nature of the motion of a bogie over square joints and staggered joints differ fundamentally. When the joints are laid square, there are two low spots opposite each other, and the bogie is free to pitch about the centre pivot. The joints (and the rolling stock) thus receive the full force of the axle load travelling into the longitudinal depression. But when the joints are staggered, there is only one low spot at a given instant, which causes the bogie to wobble — a compound motion consisting of a side and vertical oscillation — and so far as the vertical impact is concerned, the effect is much less than even that of a wheel load; for when only one wheel out of four is travelling into a depression, the three remaining wheels have a restraining influence over their truant companion, which reduces the vertical velocity of impact. Considering that the impact effect is conditioned by the square of the speed (MV_2), the proportionate reduction is much less than even that of a wheel load (as opposed to an axle load with square joints), because of the considerable reduction in the vertical speed factor.

A further point is that in normal track maintenance, very great attention is paid to the maintenance of cross-levels. When the joints are staggered, this automatically ensures a smaller difference of levels at the joints, as compared with square joints, and this correspondingly reduces the impact effect due to the lowness of the joints.

7. *Prima facie*, therefore, it can be concluded that the staggering of rail joints is beneficial, both on curves and on the straight :

(*a*) in producing better lateral continuity of track;

(*b*) in producing better vertical continuity of track; and

(*c*) in reducing the impact on the formation and on rolling stock to much less than half.

8. In the past, it was the practice to stagger joints only on curves and to employ the « mid-stagger » (half the rail length). While this arrangement secured a smooth curve, the occurrence of impacts at regular intervals of half a rail length on alternate rails was found to set up an undesirable rolling motion in locomotives and rolling stock as, in many cases, the periodicity of the roll synchronised with the periodicity of the vibration of the rolling stock springs. Pictorially the impact effect of a bogie passing over a track with mid-stagger, is shown below :

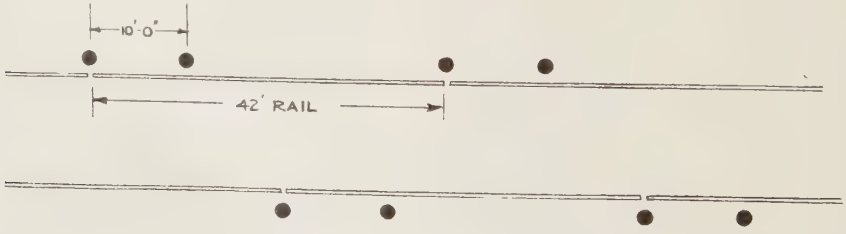


Fig. 1.

Note. — Large dots are intended to represent the timing of impacts on each rail when a bogie passes over a rail joint.

Observe that two successive impacts occur on one side of the bogie, followed by two successive impacts on the other side. The regular intervals at which these occur produces a cumulative effect which tends to build up the oscillation and increase the amplitude of the roll, causing uncomfortable riding.

9. To overcome this, the obvious solution is to reduce the pitch of the stagger to something less than the bogie wheel base, when successive impacts will occur on opposite sides of the bogie, as depicted diagrammatically below :

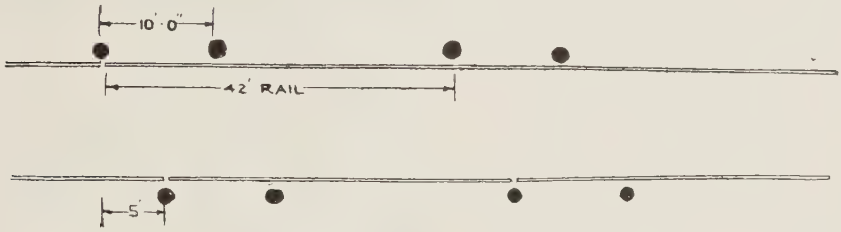


Fig. 2.

Observe that successive impacts occur on opposite sides of the bogie. This produces a quick « wriggle » or a « shudder » followed by a pause, as opposed to the roll with the mid-stagger or the pitching with square joints. The period of frequency of the wriggle is much smaller than the period of vibration of rolling stock springs, thereby eliminating the hazard of resonance. The pause following the wriggle eliminates any cumulative effect tending to build up oscillations.

10. It is more conducive to comfort if rhythm is introduced into the wriggle. For this purpose the timing of the four impacts in the passage of the bogie over the two joints should be equal. This can be achieved by adjusting the pitch of the stagger to half the bogie wheel base. It therefore follows that the optimum short-stagger is *half the normal passenger bogie wheel base,* provided that this is less than the minimum bogie wheel base of any vehicle running on the railway. Pictorially the effect produced is as shown below :

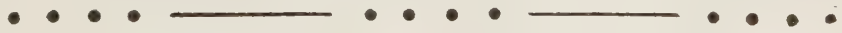


Fig. 3.

11. On the broad gauge the normal passenger wheel base is 10′-0″. It therefore follows that the optimum short-stagger is 5′-0″. As this is less than the minimum wheel base of 6′-0″ stipulated in the Schedule of Dimensions, the condition that the pitch of the stagger should be less than the bogie wheel base, is also satisfied. This is the reason why the standard short-stagger adopted on the G. I. P. Railway for relayings and block renewals is 5′-0″, with a tolerance of ± 0′-6″ to fit in with the joint and shoulder sleeper spacing and design of the composite sleeper at the rail joint.

12. For the past three years, this short-stagger of 5′-0″ has been adopted on all relayings on the G. I. P. Railway, and the reports on the running as well as the Track Recorder Charts, indicate that this disposition of slightly staggered rail joints, gives better results, both from the point of view of smooth running and improved joint maintenance, as compared with track laid with square joints or any other pitch of stagger.

2. Rail joint support with slightly-staggered joints.

The bifurcated tie-bar.

13. The aim of the sleeper spacing is to ensure, as far as possible, a uniform modulus of track support along the rail. This is achieved to some extent by increasing the density of sleepering in the vicinity of the joint to counteract the inherent weakness due to discontinuity at the joint, which is not restored to the full rail strength by the use of fish-plates.

14. It has been found in practice that with a suspended joint, the optimum joint support is secured by spacing the two joint sleepers as close as is consistent with efficient packing. The staggering of joints introduces the problem of designing a sleeper which provides an equivalent of two closely spaced sleepers under the rail joint, but only a single sleeper at the opposite end under the continuous rail. In the January 1947 issue of the *Quarterly Technical Bulletin,* the author described how this problem was solved for cast-iron sleepers by the design of a bifurcated tie-bar connected to three plates in substitution of the normal tie-bar connecting two plates.

15. Trials with this type of composite joint sleeper over a period of three years have indicated that the design was entirely sound from the point of view of alignment and gauge-holding qualities, but the close spacing of the plates under the rail joint at 14″ centres, created difficulty in packing these bowl-shaped CST 9 plates, as it permits beater packing only from the outside. This difficulty has been overcome by widening the fork of the bifurcation to permit the plates to be spaced at 18″ centres, which allows adequate space between the plate for beater packing from the inside as well. This design has proved entirely satisfactory and has been standardised for 90 lb. R. F. F. track. The arrangement is as shown below :

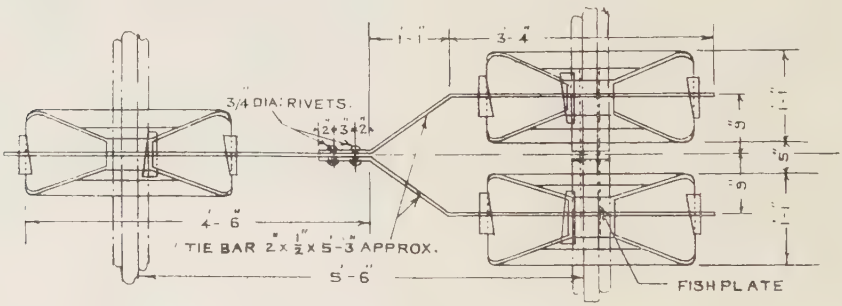


FIG. 4

Modified Duplex sleeper.

16. With a suspended joint the effect of the inherent weakness in the rail joint is to cause increased pressure in the form of a rocking action on the receiving joint sleeper with the passage of each wheel load, as the rail joint is eccentric to the joint sleeper and the load transmission, therefore, occurs through a cantilever arm. This is aggravated by the actual impact at the end of the cantilever arm. The net effect of this is to cause a permanent depression in the ballast and loosening of the packing, which is the reason why joint sleepers require more attention. If the joint is supported centrally on one continuous joint sleeper instead of two, the cantilever action and rocking effect are eliminated. The impact effect is also distributed over the whole sleeper and, if a sufficiently wide area is provided, the

effect of discontinuity as well as impact in loosening the packing and causing depressions, is minimised.

17. The precession waves of the rail formed by the motion of a vehicle on the track also cause the sleepers to be lifted and pounded on the ballast cushion. When this cushion is already depressed due to the rocking effect at joint sleepers and the packing has been loosened due to the joint impact, a tendency towards blowing is produced. If the rail is free to lift without lifting the sleeper with it, the precession waves cause no disturbance to the joint sleeper, and the final result is adequate joint support and inhibition of the tendency towards blowing at joints. This is the principle of the « Duplex » sleeper — a rail-free sleeper with a wide bearing area supporting the joint centrally.

18. An unwelcome feature, however, is that the supported joint gives rise to the problem of rail batter and high concentration of stress at the rail seat. This is sought to be overcome by providing a resilient pad to absorb the impact effect and a wide rail seating to reduce the concentration of stress. The latest modified design of the Duplex sleeper with a wooden pad and a wide rail seating conforms to the above stipulations and has, in practice, given promising results in the electrified area of the G. I. P. Railway, where there is dense traffic at normal speeds of 65 m.p.h.

19. When the track is laid to a short-stagger, it is possible to secure a uniform modulus of track support by the use of a Duplex plate under the rail joint and an ordinary CST 9 plate at the opposite end with a suitable connecting tie-bar. The arrangement is shown below :

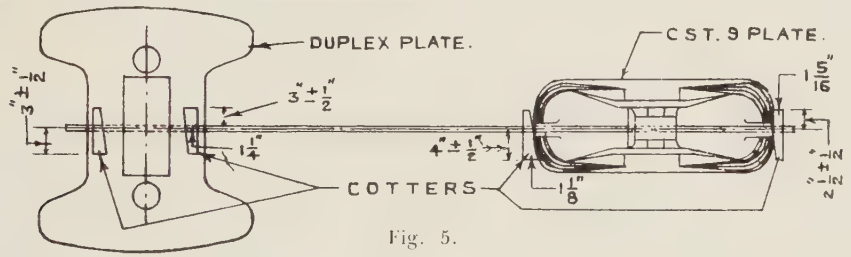


Fig. 5.

This can be used as an alternative to the composite CST 9 sleeper with a bifurcated tie-bar. The arrangement is also applicable to Bull-headed rails laid on pot or plate sleepers. It is hoped that the better support at joints will mitigate the shortcomings due to the slight variation of gauge incidental to this arrangement.

20. In a trough sleeper road, it is possible to cater for the short-stagger by using a half trough to which a tie-bar is rivetted and joggled to be threaded through a Duplex plate, as shown below :

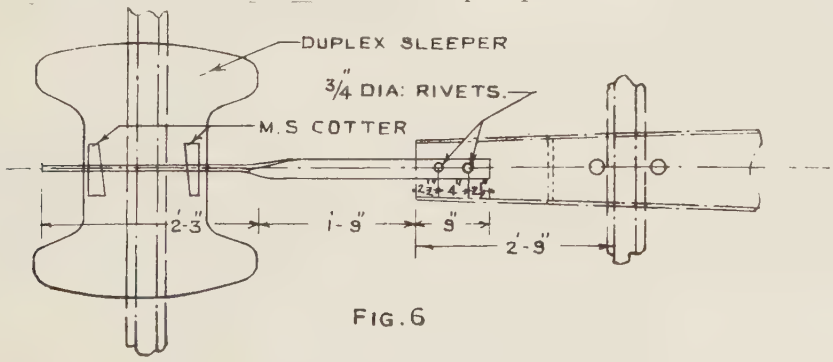
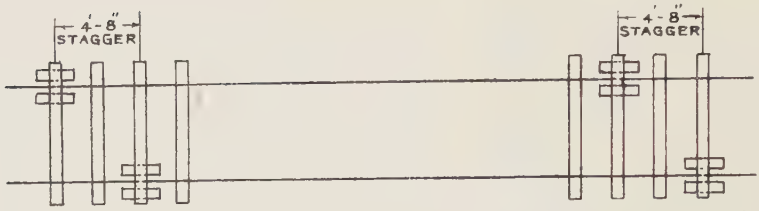
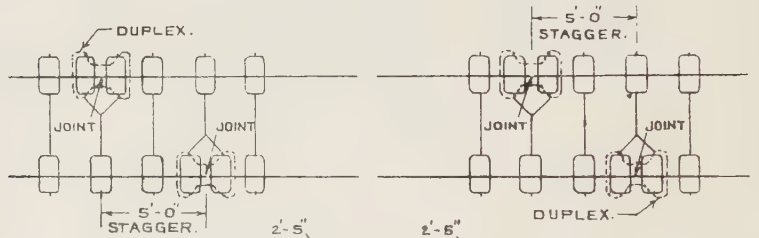
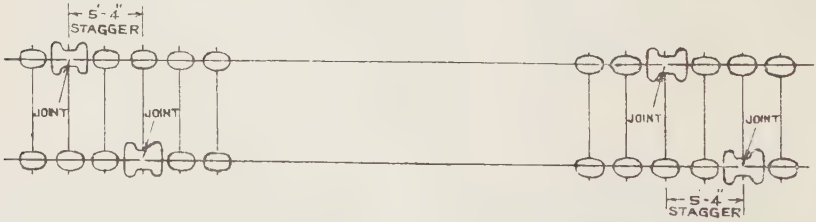


FIG. 6


4'-8" STAGGER
4'-8" STAGGER
14 SPACINGS OF 2'-4"
42' RAIL 17 WOOD SLEEPERS, 1 WOOD DUPLEX.
5 SPACINGS OF 2'-4"
39' RAIL 16 — " — " —
8 SPACINGS OF 2'-3"
36' RAIL 15 — " — " —
4 SPACINGS OF 2'-5"
33' RAIL 13 — " — " —
5 SPACINGS OF 2'-4"
30' RAIL 12 — " — " —


DUPLEX.
JOINT
JOINT
5'-0" STAGGER.
5'-0" STAGGER.
JOINT
JOINT
DUPLEX.
2'-5"
2'-6"
7 SPACINGS OF 2'-6"
42' RAIL (18 PLATE SLEEPERS) OR 16 PLATES & 1 DUPLEX.
12 SPACINGS OF 2'-5"
39' RAIL (17 PLATE SLEEPERS) OR 15 PLATES & 1 DUPLEX.
4 SPACINGS OF 2'-7"
4 SPACINGS OF 2'-7"
36' RAIL (15 PLATE SLEEPERS) OR 13 PLATES & 1 DUPLEX.
3 SPACINGS OF 2'-6½"
3 SPACINGS OF 2'-6½"
33' RAIL (14 PLATE SLEEPERS) OR 12 PLATES & 1 DUPLEX.
8 SPACINGS OF 2'-6"
30' RAIL (13 PLATE SLEEPERS) OR 11 PLATES & 1 DUPLEX.


5'-4" STAGGER
JOINT
JOINT
JOINT
JOINT
5'-4" STAGGER
42' RAIL (14 OVAL POTS & 2 POT-CUM-DUPLEX)
39' RAIL (12 OVAL POTS & 2 POT-CUM-DUPLEX)
3 EQUAL SPACINGS OF APPROX. 2'-11"
36' RAIL (11 OVAL POTS & 2 POT-CUM-DUPLEX)
6 SPACINGS OF 2'-10"
33' RAIL (10 OVAL POTS & 2 POT-CUM-DUPLEX)
30' RAIL (9 OVAL POTS & 2 POT-CUM-DUPLEX)

This device secures a uniform modulus of track support at the continuous rail at one end and the rail joint at the opposite end. It is an alternative to the 18″ sleeper spacing at joints which has also given good results.

21. In a wooden sleeper road, trials are being made of a composite wooden sleeper consisting of an ordinary main line wooden sleeper provided with two « fins » of old sleeper offcuts to give the requisite extra bearing area under the rail joint. The arrangement is as shown below :

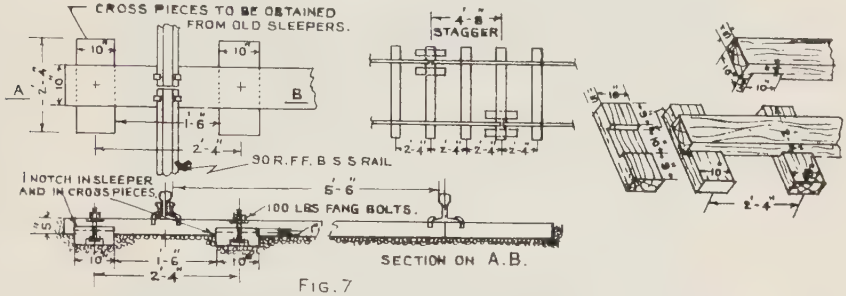

CROSS PIECES TO BE OBTAINED FROM OLD SLEEPERS.
STAGGER
90 R.F.F. B S S RAIL
NOTCH IN SLEEPER AND IN CROSS PIECES
100 LBS FANG BOLTS.
SECTION ON A.B.

Fig. 7

Up to date results of the trials with this type of composite wooden « Duplex » sleeper are promising, provided creep is held.

22. These designs of composite joint sleepers evolved for use with track laid to short-stagger on the G. I. P. Railway may be of interest to other railways who are using or experimenting with slightly-staggered joints. The keynote of all the designs is simplicity and cheapness, keeping in mind the essential purpose in view. The entire fabrication can be carried out in a small workshop with its normal resources and with indigenous material. The ideas put forward are with the object of achieving improved track standards which will also automatically ensure a higher standard of comfort for the travelling public, without recourse to the use of expensive patented devices or imported track fastenings.

23. A typical chart of standard sleeper spacings adopted on the G. I. P. Railway, with track laid to a short-stagger and with composite sleepers at joints, forms an accompaniment to this article, so as to be readily available for reference and use in trials by interested railways.

[656 .212 .8]

New features in the construction of bascule weighbridges for wagons and road vehicles.

The construction, in Switzerland, of bascule weighbridges has been developed to a remarkable extent during latter years. To reduce the time taken in weighing, modern bascule weighbridges are built without any locating winch or stop blocks, and are therefore always free for weighing or running over. Whilst the platforms of older types were usually of steel, covered with oak planks, ribbed reinforced concrete is usually preferred nowadays. The concrete bridge affords a good degree of protection to the levers against water and is itself less subject than a wooden covered bridge to weather conditions. The former triangular levers have, in addition, been replaced by individual levers which are not operated in torsion.

Figure 1 is a general view of a bascule bridge for road vehicles, whilst figure 2 shews a bridge for rail vehicles.

To facilitate access to the bridge, four removeable ramps are provided (fig. 3). These have a double purpose : passage over the rail breaks without shock, at the same time locking the platform to prevent, as far as possible, oscillation of the bridge during the passage of wheels. As the ramps can be removed, the bridge has no contact with the foundations during weighing.

Fig. 1. — Bascule weighbridge for road vehicles, weighing capacity 20 t, with ribbed reinforced concrete bridge, torsionless lever train, patent interchangeable and adjustable knife edges.

The ribbed reinforced concrete bridge rests on ball bearings, which are protected from water (fig. 5). When a vehicle runs over the bridge, longitudinally or transversely, these bearings allow a limited oscillation of the bridge in any direction. To control and damp longitudinal and transverse oscillation, adjustable shock-absorbers are fitted at each side of the bridge.

The train of levers (fig. 4) of the new type of bascule bridge comprises seven

Fig. 2. — Bascule weighbridge for wagons, weighing capacity 40 t, load capacity 110 t, allowing passage of all locomotives. Bridge of ribbed reinforced concrete; torsionless lever train, patent interchangeable and adjustable knife edges.

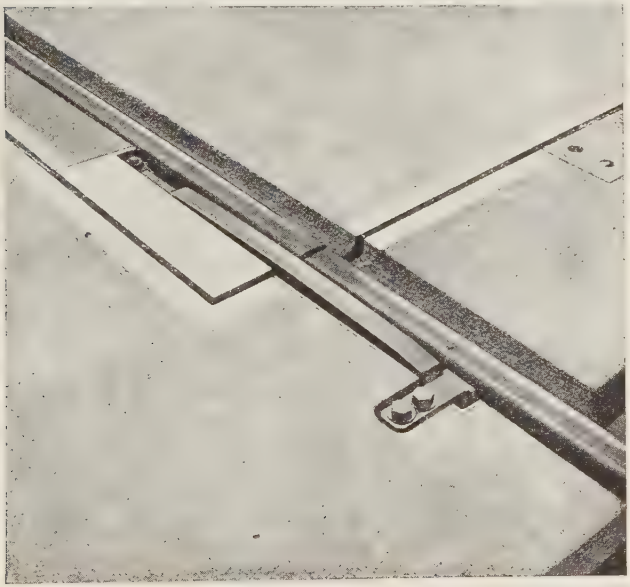


Fig. 3. — Removable ramps for bascule rail bridge.

individual light levers, which are subject only to flexing and not to torsion. The levers are fitted with a patent, interchangeable, adjustable, knife-edge (fig. 5). Adjustment and removal of these parts, which are manufactured of high-quality alloy steel, is very simply carried out by removal of two keys.

The high-ratio multiplication of the levers allows the use of a smaller balance arm and cursor weight. To avoid vibration of the balance arm during the passage of vehicles, it is kept bolted in position by a new locking arrangement, comprising a patent sprung roller.

The cursor-weight is fitted with a simple ticket-printer for impression of gross and tare weights. It is also provided with automatic protection of the notches.

If the balance arm is located in the open air, it is protected by a steel plate cover with a door and lock.

The rib construction provides much easier access for repair and inspection (fig. 6). The train of levers can be completely dismantled and taken out of the pit without any necessity for lifting the concrete bridge more than 1 or 2 mm. Any necessary repair or inspection of the bascule bridge can therefore be effected without interfering with the traffic.

Before being put into use, the bascule bridges are subject to a careful check by the Weights and Measures Office. Moreover, the Railway Company carries out loading, free running and braking tests on the bridge. Bascule bridges delivered during recent years have provided valuable experience so that newly-constructed ones give satisfaction from all points of view. All main parts are manufactured to standards and are therefore obtainable from stock.

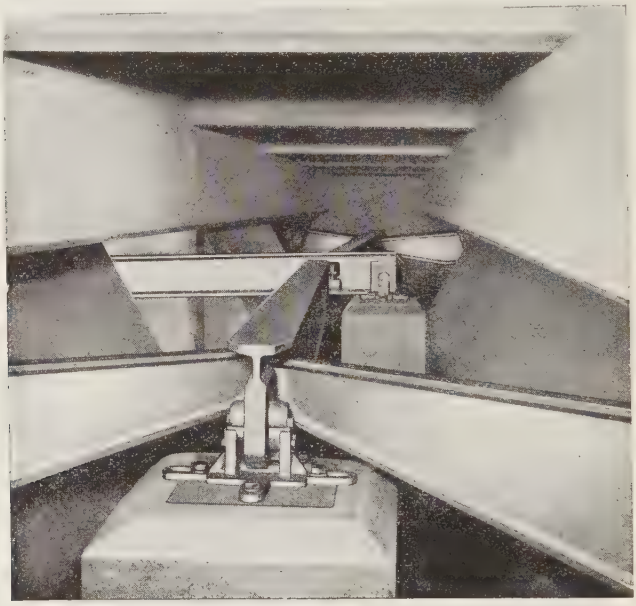


Fig. 4. — Torsionless lever train and ribbed reinforced concrete bridge for weighing wagons, weighing capacity 70 t, load capacity 120 t.

The accuracy and sensitiveness of these bascule bridges is noteworthy (fig. 7). The regulations of the Swiss Federal Bureau of Weights and Measures reads :

Up to 1/10 of the maximum load, the minimum required sensitiveness is 1/1 000; increasing the load, the 1/1 000 will remain

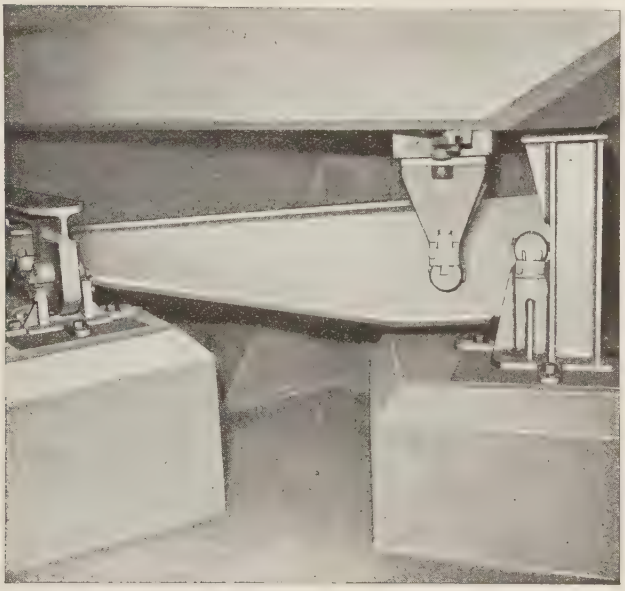


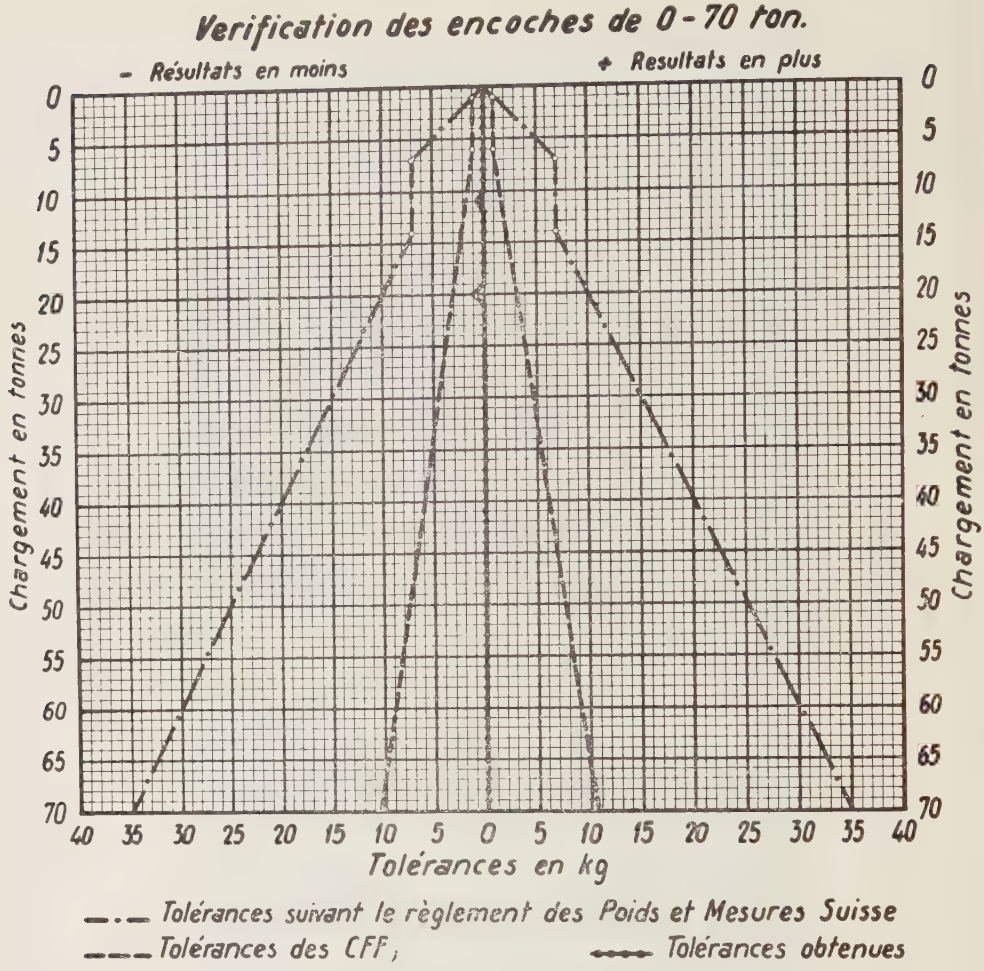
Fig. 5. — Ball bearing support with angle lever and patent adjustable and interchangeable knife edges of a 70 t bascule bridge, load capacity 120 t.

Fig. 6. — Bascule weighbridge for wagons, weighing capacity 70 t, maximum load 120 t; total platform length 11.6 m allowing passage of all locomotives. Platform of ribbed reinforced concrete.

valid until 1/2 000 of the imposed load gives a figure higher than the allowance calculated on 1/10 of the load.

Taking a bascule bridge of 70 t capacity, the tolerance limits for sensitivity and accuracy would be :

7 to 14 t : 7 kg and
to 70 t : 35 kg.


Verification des encoches de 0-70 ton.
- Résultats en moins
+ Résultats en plus
Chargement en tonnes
0 5 10 15 20 25 30 35 40 45 50 55 60 65 70
40 35 30 25 20 15 10 5 0 5 10 15 20 25 30 35 40
Tolérances en kg
Tolérances suivant le règlement des Poids et Mesures Suisse
Tolérances des CFF;
Tolérances obtenues

C 27

Fig. 7. — Tolerance ranges and comparative error curve of 70 t weighing capacity bascule weighbridge.

Explanation of French terms:

Vérification des encoches de 0-70 ton = test of notches 0-70 tons. — Résultats en moins = under recorded. — Résultats en plus = over recorded. — Chargement en tonnes = load in tons. — Tolérances en kg = tolerances in kg.

— - — Tolérances suivant le règlement des Poids et Mesures Suisse = tolerances allowed by Swiss Weights and Measures Office.

— — — Tolérances des CFF = tolerances allowed by CFF.

〰 Tolérances obtenues = tolerances obtained.

According to the latest regulations of the Swiss Federal Railways, the tolerance range for sensitivity and accuracy is fixed at 1 kg for 6 000 kg load and 0.15 $^{0}/_{00}$ for all loads above 6 000 kg; i.e. 10.5 kg for a 70 000 kg load.

The newly-constructed bascule bridges referred to in this article provide much

better results, however, and the variations are generally between 0 and 3 kg. It may be concluded that the tolerance limits for sensitivity and accuracy are improved 10 or 20 times over the Weights and Measures limits and by 3 to 5 times those of the C. F. F.

Checks of bascule bridges have shewn that variations did not exceed ± 1 kg at all stages up to maximum load. Figure 7 shews the tolerance range and comparative error curve of a 70 t bascule bridge, platform length 11.60 m (38 ft.) and maximum load 120 t.

It may be noted that, with or without load, these bascule bridges record weight additions of 0.5 to 1 kg.

As regards stability of this type of construction, it may be stated that braking and running trials with a 118 t locomotive at a speed of around 40 km/h (24 m.p.h.) on a bascule bridge have caused no variation in the tare.

The first load and running tests are normally for slight variations which come regularly between 0 and 3 kg. These good qualities of bascule bridges are maintained even after a long period of service.

Experience during latter years has led the C. F. F. and Swiss industry to prefer more and more these new constructions.

The bascule bridges described are built by the « Fabrique de Machines Louis Giroud, S. A. », of Olten (Switzerland).

[625 .232 (73)]

Burlington buys gallery suburban cars.

(From Railway Age, October 21, 1950.)

The Chicago, Burlington & Quincy is receiving a lot of 30 gallery-type suburban cars which are being built by the Budd Company, Philadelphia, Pa. The cars are of two arrangements: 25 are without toilets and seat 148 passengers each of the other five cars is provided with a toilet room and the seating capacity is 145 passengers.

The cars are built of stainless steel, assembled by the Budd Shotweld process. The construction follows the current standard of the builder in which the side structure serves as the shear members and the roof and floor as the chord members of a modified plate girder. Center sills, floor and roof are of stainless steel. The end underframe units, including draft-sill extensions, and the body bolster are of arc-welded alloy steel, completed with an alloy cast-steel end sill and coupler-carrier support. There are no skirts below the side sills, except where they form a smooth, curved connection between the vestibule step well and the side of the car.

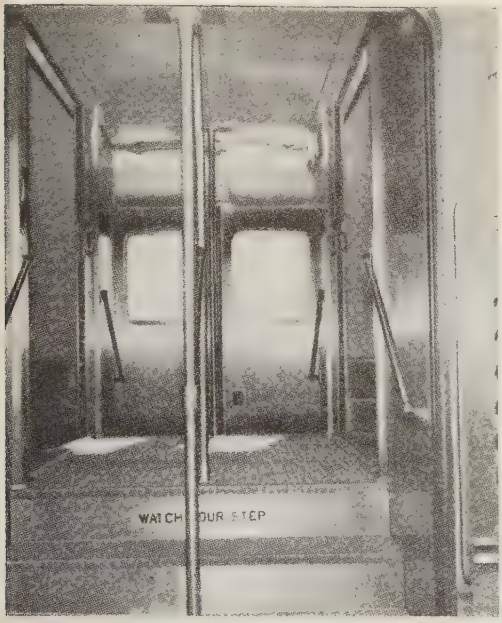
Vestibule and steps.

The vestibule and steps are in the middle of the car body, from which passengers enter either half of the car. Passengers are seated on two levels, with double seats on the main floor and single seats in the galleries. Compact stairways on each side of the car adjoin both sides of the center vestibule, leading from the aisle entrance to the galleries over the main-floor seats. A passageway, protected by a railing, extends alongside the single seats in each gallery.

The car floors in the seating space are of plywood. The lower floor is 1/2 in. water-resistant material, secured directly by blind rivets to the tops of the flanges of the stainless-steel subfloor structure. The gallery floors are 1 in. plywood, faced on both sides with metal. The floors on both levels are covered with 3/16 in. rubber tiles. The vestibule floor and steps, the gallery stair steps and landings, and the floor inside the car up to the partition at the first seats are made of stainless-steel anti-slip plates.

(Photo courtesy Hedrich-Blessing Studio.)
Looking toward the end of the car.

The gallery floor is supported at the side wall by brackets welded to the side posts and on the aisle side by truss members which serve also as the hand rail. Columns extend up from the truss panel points and are secured at the top to the roof structure to give lateral stiffness to the railings. The



The vestibule. Side doors are operated pneumatically, the end doors manually.

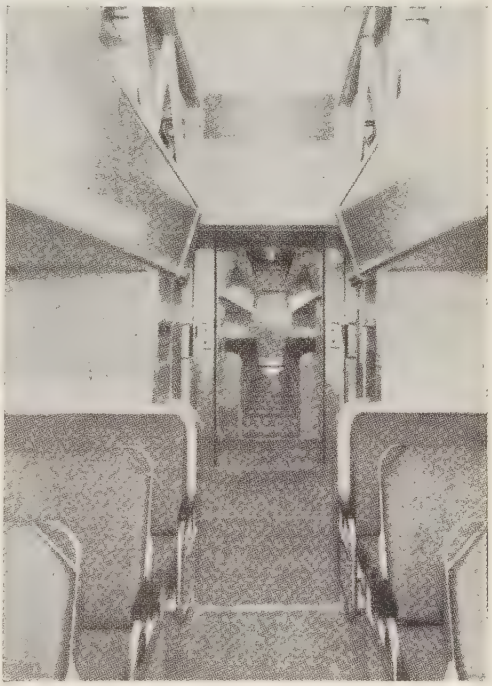
gallery structure is thus supported entirely from the side of the car and from the railing trusswork that is supported by the partitions at each end of the gallery.

All insulation in the car bodies is Ultralite, 3 in. thick throughout in roofs, sides, ends and floor. Asbestos paper is applied on the under side of the roof insulation. The insides of the side and end walls of the passenger compartments are covered with sound-deadened aluminum. Aluminum is also used on the ceiling of the passenger occupancy sections of the car.

There are two rows of six wide windows on each side of each half of the car, the upper row opening to the gallery and the lower to the main floor. Windows are of the double-glazed breather type with 1/4 in. laminated safety glass inside and 1/4 in. polished plate glass outside. Window shades are of the cable type with roller curtains.

The step wells are of the fixed type and are closed at the outside of the car by bi-parting sliding doors which extend down to the tread of the bottom step. These are operated by National Pneumatic door engines from switches in a box attached to the vestibule wall above the stanchion just inside the passenger entrance opening. These switches are arranged so that the buttons control the opening and closing of a group of cars from a single station.

Body end doors are also of the sliding type and are fitted with weather stripping. The inside aisle doors between the center vestibule and each passenger compartment on the lower floor are bi-parting sliding doors. These are manually operated and are provided with holders to retain the doors in the open position. The doors remain closed without a latch.



Looking through the vestibule. Air-conditioning evaporators and blowers are located above the vestibule ceiling.

There are 12 walkover type double seats along each side of the aisle in each half of the car. In each gallery are 11 single seats similar to the double seats on the main floor, except that there is no arm rest on the aisle side. At the end of each gallery against the end wall of the car is a fixed double seat. Seat cushions are of hair and spring construction, with padded back cushions. Upholstery is of plastic material.

above the ceiling over the center entrance vestibule.

Fresh air enters the car through four screened openings in the sides of the roof above the entrance doors. It passes through two Farr filters inside each opening before entering the plenum chamber. Each blower has a capacity of 2 400 cu. ft. per min., of which 600 cu. ft. is fresh air. The air is discharged into the upper level from the central overhead duct. It passes through

Burlington stainless steel gallery-type suburban car.

There are no parcel racks above the double seats on the main floor of the car. Under three of the gallery stairs in each car, however, is a space approximately 18 in. by 30 in. for luggage shelves. The control and switch lockers are housed in the similar space under the fourth stairway. In the gallery continuous closed-type parcel racks are placed along each side of the air distribution duct over the center aisles.

Air conditioning and heating.

Each car is equipped with two Waukesha propanedriven ice engines and two overhead Trane evaporators. Each overhead unit has a capacity of 6 1/2 tons and is equipped with supplementary heating coil and blower. The heating coil is removable without disturbing the cooling coil. The evaporator and blower units are located

branch ducts at each end of each compartment to longitudinal ducts underneath the gallery floor, from which it is discharged through slotted openings. Air is forced out of the car through two grills in the car sides at each end which discharge through two static-type ventilators.

The cars are heated by fin-tube floor radiators along the sides on both levels, controlled by Vapor single-tube thermostats. The cooling thermostats have three settings: 71-73 deg.; 74-76 deg., and 78-80 deg.

Electrical equipment.

These cars receive their electric power supply from generators mounted on a power car directly behind the locomotive. Distribution is through a loop system train line of three wires, suitable for transmission of 61-V D.C.

The battery is an Exide type MVAHT-13 of 213 amp.-h. nominal capacity at the eight-hour discharge rate. It is used only for ice-engine starting and layover heat control. All lighting is incandescent. The lighting of the passenger space in the car is by center ceiling lights and by reading lights over each seat.

In the gallery. Parcel racks are under the ceiling between the galleries.

Four-wheel trucks.

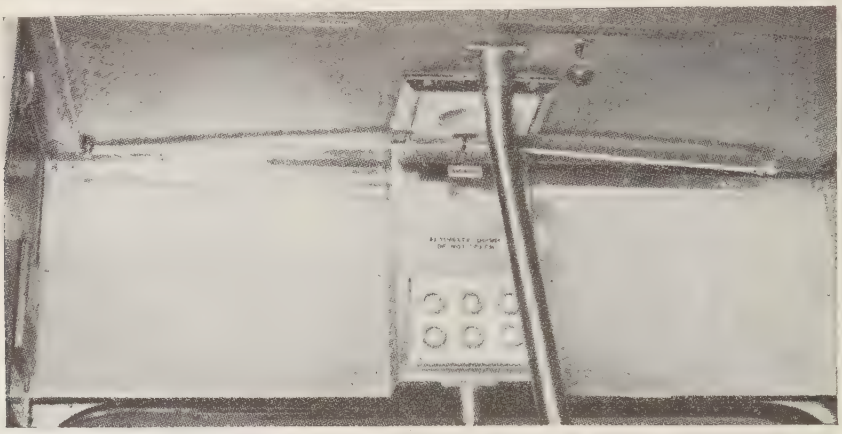
The cars are carried on four-wheel trucks with a wheel base of 8 ft. 6 in. Frames and bolsters are alloy-steel castings. The single equalizers are forged. Bolsters have roll stabilizers and longitudinal anchor rods. All springs are helical and Houdaille shock absorbers are applied at the ends of the bolsters. Journals are 6 in. by 11 in., fitted with Timken roller bearings and wheels 33 in. in diameter of high-carbon rolled steel. Treads are semi-cylindrical. Rubber pads are placed between the center plates, under the side bearings and at the ends of the bolsters. Rubberized fabric pads are placed between the ends of the equalizers and the tops of the journal boxes.

The trucks have clasp brakes of the Unit-cylinder type. A lever type hand brake mounted on a collision post at one end of the car operates through the clasp brakes on the one truck adjacent to the brake.

The air brakes are of the modified H S. C. type with pneumatic control, furnished by Westinghouse. The braking ratio, based on the ready-to-run weight, is 150 % at 100 lb. cylinder pressure. Provision is made for the later application of the straight-air pipe.

In order to permit operation of the angle cock without going under the end of the car, there is a loop in the end of the brake pipe. In this loop is located the angle cock, thus bringing it close to the side of the car.

There are four schemes of interior decoration. All employ tan and cream for walls and ceilings, and rust and green window shades. In two, the seat covering is plastic upholstery in green and the floor tiles green and ivory, or rust and ivory mar-



Buttons for opening and closing the vestibule side doors.

The cars are fitted with tightlock couplers and Waughmat draft gears. The ends of the cars have face plates of stainless steel and single-fold inner diaphragms. A canvas boot at the bottom of the diaphragm aids in closing the openings at each side of the foot plate.

bleized. The others have plastic upholstery in rust and rust and ivory, or green and ivory floor tiles.

The cars are 85 ft. in length, coupled, have a maximum height above the rail of 15 ft. 8 in., and weigh 132 380 lbs., each, ready to run.

[625 .213]

Formula for the critical buckling load of a helical spring.(*)

In an article in the July 1947 *Bulletin* (1), the author, Mr. KOFFMAN, gave the following formula for the critical buckling load of a helical spring :

$$P_c = \frac{B\frac{\pi^2}{L^2}}{1 + \frac{B\pi^2}{SL^2}} \quad . \qquad (1)$$

This formula is only suitable for small values of the ratio r/L : the exact formula for the critical load is given by HARINGX(2)

We will compare briefly the two formulae for the case of circular section rod. Using the nomenclature of Mr. KOFFMAN and introducing the relative compression

$$\xi = \frac{P_c}{\gamma_o} = \frac{E}{GS} = \frac{2\,(m+1)}{m\,S} \qquad (2)$$

(m being the constant of POISSON and γ_o the rigidity for the pressure per unit of length of the spring free) the exact relation between r/L and ξ can be written :

$$\frac{m}{2\,m+1}\,\pi^2\left(\frac{r}{L}\right)^2 = \frac{m}{2\,(m+1)}\,\xi\left\{1 - \frac{m+2}{2\,(m+1)}\,\xi\right\}. \qquad (3)$$

whereas formula (1) gives for this relation:

$$\frac{\frac{m}{2\,m+1}\,\pi^2\left(\frac{r}{L}\right)^2}{1 + \frac{m}{2\,m+1}\,\pi^2\left(\frac{r}{L}\right)^2} = \frac{m}{2\,(m+1)}\,\xi. \qquad (4)$$

It will be noted that these formulae become the same for large values of r/L and low values of ξ.

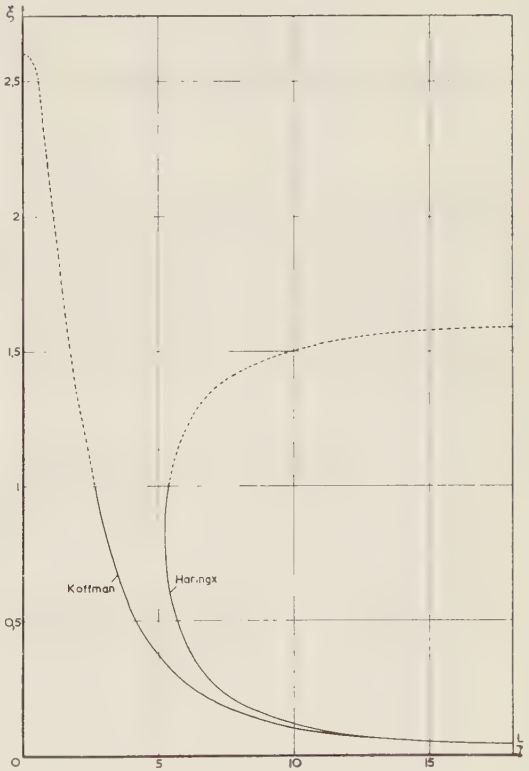


Fig. 1. — The relative compression as a function of the ratio $\frac{L}{r}$.

(*) Note sent by Dr. A. D. DE PATER, Netherlands Railways, Utrecht.

(1) J. L. KOFFMAN, « Light Railcars for Colonial Railways », *International Railway Congress Bulletin*, July 1947, p. 669, formula 17.

(2) J. A. HARINGX, « On highly compressible helical springs and rubber rods, and their application for vibration-free mountings », *Philips Research Reports* 3 (1949), p. 411.

For when $m = \frac{1Q}{3}$ we have calculated the curves in the plan $\left(\frac{L}{r}, \xi\right)$ indicated by the equations (3) and (4) : see figure 1. For the values $\xi > 1$ (1) the curves are dotted.

From this graph we can deduce that the Koffman formula gives a critical load P_c for all ratios of $\frac{L}{r}$, whereas the Haringx formula shows that such a force does not exist for $L/r < 5.24$. This latter result has been confirmed experimentally.

It is therefore desirable to use the Haringx formula in preference to that of Koffman especially as there is no difference as regards the amount of calculation to be done. In our opinion, the Koffman calculations according to the formula (17*a*) should only be applied with due reservations.

A. D. DE PATER.

(1) J. A. HARINGX has shown that these same values have a practical importance, contrarily to what one would be inclined to believe.

[385. (09 .2]

OBITUARY.

Marcel CASTIAU,

Honorary General Secretary of the Belgian Ministry of Communications.
Managing Director of the National Office for the completion of the « Nord-Midi » Junction.
Former Member of the Permanent Commission of the International Railway Congress Association.

On the 16th September 1950, there passed away one of the most outstanding personalities of the Belgian Railways, and a devoted friend of our Association : Mr. Marcel Castiau, former Chief Engineer, Manager of the Special Works and Research Department of the Belgian Railways, Honorary General Secretary of the Ministry of Communications, and Managing Director of the National Office for the completion of the « Nord-Midi » Junction.

Mr. Marcel Castiau was born at Renaix, on the 28th September 1877. He had a brilliant scholastic career at the Atheneum and then at the University of Ghent, where he gained a diploma in Civil Engineering.

He entered the service of the railway on the 1st June 1901 after a competitive examination in which he was top of the list. He was at first employed by the Management of the Special Works and Research Department of which a well known engineer Mr. Van Bogaert was then the head; the latter appreciated from the start the rare qualities of his young assistant, whom he held in great estime.

His main duties were in connection with important civil engineering schemes, bridges, metal bridges, the layout of stations, buildings, etc.

In 1902, he was charged with studying the construction of a railway in Asiatic Turkey, from Damas to Mecca; in 1906, he was laying out a new railway line in Colombia, and in 1908 we find him in Haïti, where the revolution surprised him.

The University of Brussels engaged him from 1910 to 1914, and he was in charge of the course on the construction of bridges.

In 1917, he was appointed Chief Engineer of the Special Works and Research Department of the Belgian Railways.

From the 1st June 1919 to the 10th December 1920, Marcel CASTIAU was employed by the Auxiliary Railway Company of Brazil, where he acted as General Manager of a Brazilian Railway (about 2 300 km = 1 430 miles of lines).

From the 24th February 1921 to the 29th March 1924, he was in the service of the Chinese State Railways, on the Lunghaï to Pien-Lo lines.

On the 25th June 1925, Marcel CASTIAU was appointed Head of the Cabinet of Minister Anseele.

When the Belgian National Railways were set up, Mr. CASTIAU was appointed General Manager of the « Service des Chemins de fer Concédés », attached to the General Secretariat of the Railways, Marine, Posts, Telegraph, Telephone and Aeronautical Department.

Once again, he accepted a mission to Persia in the service of the Belgo-Persian Company from the 8th May 1928 to the 1st April 1929.

In October 1929, he was promoted acting General Secretary to the Ministry of Transport, and given the title of General Secretary on the 24th June 1931.

A royal decree of the 9th September 1935 appointed him as Managing Director of the National Office for the completion of the Nord-Midi Junction.

On the 1st April 1941, he was retired from his post as General Secretary by the decree dealing with age limits taken during the occupation of the country.

Put on the retired list as from the 1st October 1942, he was authorised on the 4th September 1944 to resume his duties as General Secretary for a period of 6 months.

On the 4th March 1945, he was given an honourable discharge from these duties and went into retirement.

As railway engineer, Mr. Marcel CASTIAU was responsible for the raising in level of the Brussels-Antwerp line, and that of the Brussels-Midi station and the lines running into it, as well as for the carrying out of the new line from Brussels-Midi to Ghent St. Pieter, and the diversion of the Brussels-Nord-Quartier Leopold Ceinture Railway. After the 1914-18 was, he was in charge of the work of reconstructing the railway bridges in the devastated regions.

At the National Office of the Nord-Midi Junction, Mr. CASTIAU followed very closely the technical studies prepared by his co-worker Mr. Franchimont. He was responsible more especially for the administrative part of the work, the labours of the Administrative Council, relations with the Central Administration of the Department of Communications and the other public administrations, the City of Brussels, and the different communes interested in the Junction. He carried on with this work up to the very end.

Mr. Marcel CASTIAU won a pre-eminent place as an economist and sociologist; he was the author of many studies the diversity and great interest of which bear witness to his prodigious activity in every field. Amongst others, we may recall amongst those dealing more particularly with the problem of transport, the following works: « The present and the future of Transport in Belgium » (1936); « Transport in Belgium and its Co-ordination » (1935), which were much appreciated when they made their appearance. He contributed in addition to many technical and economic reviews, both in Belgium and abroad, and his well informed impressions are recorded in several works devoted to various foreign countries where he carried out technical duties.

Mr. Castiau was elected Member of the Permanent Commission of our Association in 1927. His participation in the works of the Association dates back to the Rome Congress in 1922, at which he presented a noteworthy report on Question IV : « Reinforced concrete — the use of ordinary and reinforced concrete on the railway ». He represented the Belgian Government at the Madrid Congress in 1930 and Paris Congress in 1937, and also took part in the Enlarged Meetings of the Permanent Commission held in Brussels in 1935 and 1939.

He resigned from the Permanent Commission upon retirement in 1945.

His brilliant career was crowned by the orders conferred upon him : High Officer of the Order of Leopold and Commander of the Order of the Crown. He was also awarded many foreign orders, amongst them that of Officer of the Legion of Honour, Commander of the Order of the Polar Star (2nd class), Grand Officer of the Order of the Nile (Egypt) and Order of the Three Stars of Lettonia.

We will keep the best memory of our deeply regretted colleague, and wish to convey our very sincerest sympathy to his family.

The Executive Committee.

[385. (09 .2]

OBITUARY.

Egil SUNDT,

General Manager of the Norwegian State Railways.
Member of the Permanent Commission of the International Railway Congress Association.

We have learnt with regret of the death on the 6th September 1950 of our colleague Mr. Egil Sundt, General Manager of the Norwegian State Railways, and Member of the Permanent Commission of our Association.

Mr. Egil Sundt was born on the 23rd September 1903. After a brilliant scholastic career, he matriculated in 1922. He then studied law and passed his examinations in 1927.

He began his administrative career in 1929 as secretary to the Ministry of Finance. His brilliant qualities led to his promotion to chief clerk of the same Ministry, in 1933.

In 1939, he was appointed Director of the Norwegian Radio Service (Norsk Rikskringkasting). He left this post during the war and became Director of the Assurance Company « A/S Norske Alliance ».

At the end of the was, Mr. Egil Sundt once more took over the duties of Director of the Norwegian Radio Service.

In 1946, he was appointed General Manager of the Norwegian State Railways. During this difficult post war period, Mr. Egil Sundt was responsible for the immense task of restoring and re-equipping the Norwegian Railways.

He was overtaken by death in the very prime of life, to the great regret of all his colleagues.

Mr. Egil Sundt was appointed Member of the Permanent Commission of our Association at the Enlarged Meeting held at Lisbon in June 1949, when it was decided to create a new mandate in favour of Norway. Mr. Sundt was thus the first holder of this mandate, and he always manifested the greatest interest in the works of our Association.

We wish to offer his family our most sincere sympathy.

The Executive Committee.

NEW BOOKS AND PUBLICATIONS.

[**385** (08 (592)]

Annual Report of the Burma Railway Board 1947-1948. — One brochure (8 1/4 × 13 in.) of 64 pages with numerous tables and illustrations. — 1949, Rangoon, published by the Burma Railway Board.

Four hundred thirty thousand square kilometres (166 000 square miles) in extent, Burma was until recently a British province of western Indo-China. In the south, it is bordered on the west by the Gulf of Bengal and on the east by Siam. Further north its frontiers consist of Bengal on the left, Yunnan (China) on the right, and above Southern Tibet.

The country consists of large valleys running north and south, separated by chains of mountains ending in vast deltas.

Amongst the products of the country, rice holds the foremost place. The forests are another source of wealth. Here teak is found and is the source of an important traffic. Oil is found in the soil, and there are also coalfieds.

The backbone of the railway system from which all the other lines branch off starts at the Port of Rangoon and climbs towards the north for about 1 000 km (623 miles), first of all through the valley of the River Sittang. The lines extended to 1 777 miles in 1948 compared with 2 058 miles in 1941.

The period covered by this report is characterised by an important historic event. On the 4th January 1948, Burma became an independent sovereign Republic. Happily the new status came about without any violence, as the result of a treaty concluded with the Government of Great Britain. It has led to important modifications in the management of the railway as also in its Administration. Since the 1st January 1948, all important posts have been given to Burmese employees.

In spite of the favourable circumstances surrounding the birth of the new regime, the year 1948 was darkened by internal troubles, the countereffects of which fell heavily upon the railway.

The destruction wrought by the insurgents impeded the traffic and stultified the great effort already expended in repairing the damage done by the war.

The report gives detailed statistics for the traffic and the receipts, the state of the stock, the lines and permanent way, and an analysis of the train and engine mileage.

In the commentary given alongside these figures, the authors have stressed the progress made in reconstruction, the recovery of traffic, and the improvements made in the organisation of passenger and goods transport.

Amongst the work of reconstruction mention is made of many metal bridges, some of which are of very long span. The very great differences in the levels of the rivers added to the difficulties encountered by reason of the steel shortage, and lack of equipment and labour.

The report also describes the steps taken and the results obtained in the repair shops and the reconstitution of the rolling and traction stock. To replace the telephone and telegraph communications, it was decided to build radio stations at three important stations, but only two of these were working in 1948.

The different details given in this very interesting document show that the management had a struggle to put the railway back on a sound footing in spite of many adverse factors.

Nevertheless, the goods traffic in ton-miles exceeded that of the previous year by 20 %. A less satisfactory feature was the increase in the proportion of the low tariff traffic compared with the high tariff traffic. This led to a decrease in the average ton-mile receipts.

E. M.

[**385** (09 (436)]

SEIDLER (Dr. Ernst), General Manager of the Austrian Federal Railways. — **Fünf Jahre Wiederaufbau Osterreichische Bundesbahnen, 1945-1950** *(Five years of reconstruction on the Austrian Federal Railways, 1945-1950)*. — Special issue of the review *Eisenbahn*. — One brochure (8 1/4 × 11 3/4 in.) of 110 pages, copiously illustrated with map. — 1950, Vienna, Messrs Ployer & Co., Publishers, Ogidigasse, 5.

The Austrian Railways have undergone several transformations during the last quarter of a century. Already after the first world war, the destruction of the monarchy caused staff difficulties and financial troubles; in 1923, they were commercialised, that is to say they became an autonomous undertaking. The Anschluss of 1938 led to their incorporation in the German Reichsbahn, and resulted in a rating system which had little relationship with the mountainous character of most of the lines. Finally, the second world war after having exposed them to many trials, left them in a very difficult situation.

At the end of the war only a very reduced staff remained to cope with the material difficulties, with no organisation and no leadership. In an introductory note, the author explains how the services were reconstituted with the added complication of foreign occupation. But he insists above all upon the devotion to duty and sense of organisation of the Austrian railwaymen which was manifest during the first weeks after the liberation.

In the following pages, the author depicts the state of the railway in the spring of 1945, and notes the various stages of reconstruction. He does this by examining all the aspects under which a railway functions: management, organisation, staff, rates, receipts and expenses, road competition, international traffic, organisation of the trains and evolution of the traffic, fixed installations, rolling stock and electrification.

The restoration of the installations and the reconstitution of the stock were however not merely limited to restoring them to the pre-war level. From this point of view it is interesting to note the innovations made in the layout of the stations and their offices such as the rolling stock sheds. A considerable effort was made to improve the rolling stock as well. Road services were extended, both in the case of passenger and goods transport, and equipped with new vehicles. In this way the Austrian Railways came to the assistance of the population which had been severely tried.

A report of this kind cannot avoid excursions into the past. In this case, these enable instructive comparisons to be made with the new period, especially as regards the staff, the rates, the financial results, and the effectiveness of the methods applied. The railway however is a long way from enjoying complete liberty to enable it to cope with full success in difficult situations, especially as regards road competition, where the circumstances are very unfavourable as in other countries.

As regards electric traction, the installations suffered less than in the case of steam traction, and could be restored to service as soon as the permanent way was rebuilt. The advantages of electric traction were seen to be so great in this period of economic instability that a further electrification programme covering 2 000 km (1 242 miles) of line was investigated. After this work has been completed, half

the Austrian system will be electrified. The lines selected carry 80 % of the total traffic of the system.

The persevering efforts made in reviving the railway found their due reward in the regrowth of the traffic. The diagrams appended show that in 1949 it frequently reached and even exceeded the figures for 1937.

Out of 6 048 km (3 758 miles) of line, 2 507 km (1 600 miles) were out of action. At the present time all but 25 km (15 miles) are open to traffic again. The work accomplished in five years is very considerable. In reading this history of the restoration of a railway, in the midst of events that were often dramatic, one can appreciate once more the pre-eminent part played by the railway in the life of a nation.

E. M.

[**385** (08 (59)]

SANDERS (J. O.), C. M. G., M. Inst. T., A. M. Inst. C. E., General Manager of the Malayan Railway. — **Malayan Railway Report for the year 1948.** — One brochure (7 7/8 × 13 in.) of 72 pages, with illustrations and plates. — 1949, Kuala Lumpur, Cayton Press Limited.

This report was published in the same form as that for the previous year (see *Bulletin* for January 1951). As before it includes statistics relating to the traffic, train services, receipts and an analysis of the goods carried. A study of the costs, the general financial results and the position as regards rolling stock and installations completes the report of the evolution of the railway during 1948.

Further progress has been made in re-establishing the railway. The length of lines in operation has increased from 810 to 860 miles compared with 1 608 miles in 1939. A delivery of Diesel-electric shunting locomotives and new wagons has increased the operating possibilities. Reconstructions and improvements have taken place in the case of stations, harbours and rolling stock sheds.

Unfortunately, the year 1948 was darkened by the acts of terrorists. Bridges, the permanent way, signals and telecommunication lines were attacked. Trains were fired upon. This resulted in serious upsets to the traffic, especially in the case of the passenger services, and additional expenditure due to the defence measures which had to be taken. At night pilot trains had to be run in front of the regular trains.

In spite of these unfavourable circumstances, the receipts reached the highest level to date, exceeding those for 1947 by 12 %. The increase is due to the goods traffic, the amount of passenger traffic being much below that expected. However the costs rose still higher (23 %), mostly on account of the increase in wages and salaries which have increased by 27 %.

All these figures relate to the railway as a whole as well as the other services operated by the Malayan Federation, such as the harbours, ferrys, and road transport services.

The general tariffs did not undergo any modifications during 1948.

The report brings out two particular facts to be included in the various difficulties encountered in operating in this tropical country. The rainfall is so variable that statistics are kept of the damage caused by water to cuttings and embankments. There were 215 landslides and floods during the year. Owing to the climate, the staff suffered a great deal from malaria. The health service has had much success in this connection by increasing the work undertaken and using considerable quantities of anti-malarian oil and D. D. T. products (hexachlorocyclohexane).

E. M.

[347 .763 .4]

AUBRY (C.), Honorary General Inspector of Bridges and Highways, former Chief Engineer attached to the Permanent Way and Buildings Department of the Alsace-Lorraine Railways. — **Législation des Chemins de fer (cours abrégé)** *(Railway Legislation [short course])*. Tenth edition revised and brought up to date. — One volume (6 1/4 × 9 1/2 in.) of 190 pages. — 1949, Paris (V^e^), Editions Eyrolles, 61, boulevard Saint-Germain.

The subject matter of this book is very abundant and very varied. There would already be plenty of ground to cover if it were only a question of the relations between the railway and its clients, i.e. the whole of the legislation and regulations which cover the drawing up of the rates and constitute the essence of the transport contract. It appears however that the author wished to make his work more extensive and complete by including the official legislation affecting the railway not only in so far as it is a public transport service but also as an industry.

Moreover, if we consider that the various categories of lines are dealt with (main line railways, local railways, industrial lines), that the regime affecting the French railways has undergone profound modifications on several occasions, even quite recently, that the commercial operation of the railway has also been completely reorganised and that the co-ordination of transport has already inspired several decrees, we shall get some idea of the number of documents which had to be analysed.

Doubtless, it is on account of the abundance of the matter that the author has adopted a condensed form, under the title of short course. The work does not lose anything thereby as regards clarity nor usefulness for reference. The most important regulations and the most striking facts are brought out by appropriate though brief commentaries.

To support what we have just said and explain more clearly the different questions dealt with, we will quote the subject matter of the different chapters of the book :

I. — Preliminary (categories, regime, organisation).

II. — Present regime of the main line railways (S. N. C. F.).

III. — Notes on checking the accounts of the Companies.

IV. — Regulations concerning the operating and policing of the railways.

V. — Regulations concerning accidents.

VI. — Transport of dangerous or infected goods and livestock.

VII. — Hygiene, cleanliness and disinfecting of the different railway installations.

VIII. — The railway staff.

IX. — Idea of commercial operation.

X. — Organisation of the services controlling transport.

XI. — Local railways. Quay and harbour lines. Subsidised motor services.

XII. — Co-ordination of transport.

XIII. — Industrial lines. Brief review of colonial railways.

This list gives the impression that it would be difficult to find any text concerning the railway, which has not been studied or quoted in the book.

To be sure it was not possible to give even in part the actual text of the laws, decrees or regulations. But the precise indications given enable the reader to refer to them very easily. However the author wished to make this unneccessary to some extent by giving as an appendix the text of what he considers the most important laws. These are : the law on the policing of the railways, the law and decree con-

cerning gates and barriers, the decree of 1942 on policing and operating, and finally the regulations of the S. N. C. F. of the 31st December 1937.

A complete repertory, a knowledgeable guide, an authoritive commentary, are the attributes which come to the mind to describe this book which is published in a very handy form for study.

E. M.

[**625** .14]

ANTIA (K. F.), Chief Controller attached to the Indian Ministry of Railways, Central Standards Office. — **Railway track. Design, construction, maintenance and renewal of permanent way with notes on signalling and bridge maintenance.** — One volume (5 1/2 · 9 in.) of 416 pages, copiously illustrated. — 1949, Bombay, published by the New Book Company Ltd., 188-190, Hornby Road.

The scope of this book as explained by the subtitle is covered almost entirely by the first two parts. One of these describes the various parts of the superstructure, and the other deals with the various questions affecting their use and their maintenance in service. A third part groups together various supplementary questions, in particular the signalling and maintenance of bridges, whilst a fourth consists of a whole lot of information on various points.

Published as a guide for the permanent way staff, the book devotes much space to both theory and practice. Without sacrificing rigorous reasoning, the mathematical apparatus contains all the essentials and is limited to the extent required by the object in view. The formulae established or quoted are generally followed by concrete examples right up to the final figures.

These remarks apply in particular to the chapters dealing with the special track equipment, crossings, turn outs and switches, and their positioning in branch lines and groups of lines.

They also apply to the developments concerning the stresses to which the track is submitted under the rolling loads of the trains. This chapter in which theory is supported by experience and the measurements carried out in America and India must be mentioned together with the others to show the modern and progressive outlook of the work.

The question of curves with all their special features, over-gauge, check-rails, superelevation, transitions, speed restrictions, short rails, etc., is covered completely in that part of the book dealing with the laying and maintenance of the permanent way. In the case of rectifying curves and inserting transitions, the author gives details of a method based on measuring the versines by means of short cords, a method used successfully during recent years in India and elsewhere.

The practical utility of this book, which is manifested throughout, is above all apparent in the chapters dealing with the work of laying the permanent way and the organisation of the maintenance. All the details of carrying out the work are noted methodically and minutely.

In many cases, it was necessary to take into account the existence of various gauges, as both the permanent way and equipment has very different characteristics according to its gauge. Whenever necessary the author has never failed to give the precise details required by those who have to carry out the work.

Amongst the subjects dealt with in the third part are the manufacture and testing of permanent way materials, including the rails, fishplates and bolts, the steel and casting of metal sleepers covered in the first part of the work.

The 19 appendices grouped together in Part IV contain much information,

mostly numerical, of great practical value. Amongst these we may mention the dimensions, weight and mechanical properties of the standard rails, together with the fish-plates and bolts. A table gives the super-elevation needed for various radii of curves and the speed restrictions involved. Another gives a choice of standard dimensions relating to the Indian lines.

E. M.

[385 (09 (6)]

LARTILLEUX (H.), Former pupil of the Polytechnic School, Principal Inspector of the French National Railways. — **Universal Geography of Transports.** *Tome I* **: Geography of the French Railways.** *Volume I* **: North Africa.** — One volume (7 7/8 × 11 1/2 in.) of 194 pages, copiously illustrated with coloured maps. — 1950, Paris (IX^e), Librairie Chaix, 20, rue Bergère.

Readers of the *Bulletin* will have seen in a review published in the January 1950 issue that the first two volumes of this work described the railways of the mother country, including the *lines of the S.N.C.F.* and the *various companies.* Volume I will be completed by two others, this one describing the railways of *North Africa,* and a fourth which will deal with those of *France overseas.*

There are three chapters in this book, entitled respectively : *Algeria, Tunisia,* and *Morocco and the Sahara.* We can get a birds eye view of the whole of the communications between France and her African Colonies, by rail, sea and air, by looking at the map at the beginning of the book. In Africa, the black lines denoting the railways give place as we get into the interior of the continent to important road services linking together the different territories of the French African possessions. One of these seems to have shown the route to be followed by the Trans-saharian (Mediterranean-Niger) dealt with at the end of Chapter III.

The same general plan with the variations indicated by the actual contexture of each system has served as the basis upon which each chapter was written. The history of the railway and the alterations made to the administrative regime, operating methods, the track and the rolling stock, the layout and profile of the main lines and the pioneer lines, and finally the urban and suburban systems are the cores around which gravitate all the factors submitted for analysis.

Many maps in colour or black and white, proper geographical maps, contribute towards giving the book the character claimed by its title. In them we can appreciate the coast lines, the economic centres of activity, the bas-relief of thè country, the rivers, and all the twists and turns of the railway system. In the case of the latter detailed profiles form a real cross section of the country. The history of the railway is closely linked up with that of the development of all the fertile regions rich in the products of the soil and sub-soil, and that of the large cities, with their industrial or harbour installations. There are some very large cities : Algeria has 400 000 inhabitants, Casablanca 500 000. Around the large centres, the evolution of the railway lines and the arrangement of the stations are shown on large scale maps, which in actual fact are plans of the towns and their surroundings.

Turning over the pages of this book, one is struck by the abundance, richness and variety of the illustrations. Together with the documents where drawings abound, there are many photogravures; both together give a true picture of the local colour and recreate for us the African scene.

From the technical point of view, this book is also extremely instructive. It is remarkable to see how the characteristics of the lines linked up with the climate affect the methods of traction and types of rolling stock used. A new field is opened to electric traction and important projects have seen the day, encouraged by fruitful realisations.

The great diversity of the North African lines will be noted. There is the same diversity in the important artery known as the Imperial Line (Morocco-Algeria-Tunisia), which can be explained by the circumstances under which the different sections were built, the history and description of which are extremely interesting; here, as in many other places, considerable improvements are now being made.

The Mediterranean-Niger line is a subject well fitted to crown the book and chapter III (Morocco and the Sahara). This line of world interest is called upon to establish economic relations between Continental France and North Africa on the one hand, and West Africa and in the future Central Africa on the other hand.

The author gives its history by explaining the layout, the pecularities of construction and how it will be operated, and finally bases his estimate of the probable traffic on a very solid basis.

E. M.

[**385** (09 (495)]

The reconstruction of S. E. K. Railways. — Published by the Management of New Works of the S. E. K. (Athens). — One brochure (8 1/4 × 11 in.) of 34 pages, copiously illustrated. (Publication in English and Greek.)

In Greece, the most important railway by far is the State Railway. This includes the main line from Athens (The Pirea) to Platy (527 km = 328 miles), that from the Gulf of Salonica towards the west as far as Florina near the Jugoslavian frontier (205 km = 133 miles), a third which also starts from Salonica to Idomeni in the north (77 km = 48 miles), and finally the line from Salonica which runs through Macedonia to Alexandroupolis in the east (477 km = 296 miles). The State lines are standard gauge lines (1.435 m = 4′ 8 1/2″).

After the expulsion of the Germans all traffic became impossible. The destruction wrought by the enemy in retreat left nothing but ruins in its wake. The value of the installations destroyed was estimated at 5 200 million drachmes before the war. This disaster was all the more cruel as the S. E. K. is the backbone of land communications in Greece.

The report presented here by the New Works Department gives details of the extent of the damage suffered and the measures taken to restore the lines to traffic as quickly as possible.

Amongst the difficulties reported or guessed at must be mentioned the absence of any alternative method of transport in the mountain districts. It was also necessary to clear the minefields laid by the Italians and Germans, work which cost the lives of many railwaymen, and there was still the action of guerillas to be feared.

Begun in the spring of 1945, this work was more or less completed by the end of 1949. The services were running again everywhere except on one section of the Salonica-Florina line where there are five tunnels and many viaducts.

Although quite a time has elasped since the liberation of the country, it is impossible to consider it excessive in view of the immensity of the task to be accomplished.

The report explains the organisation of the work, the assistance given by American organisations, the main work carried out,

how the rolling stock was reconstituted, together with an estimate of the cost and the way the work was financed.

In all 11 tunnels were destroyed, that is to say every tunnel, and 615 metal bridges. To replace the latter 15 970 tons of steel were required.

The report pays tribute to the considerable aid given by the Allies and the interest taken by the Americans in the reconstruction of the railway, as well as to the devotion and spirit of sacrifice of the Greek railwaymen.

A great many drawings are given in the appendices, with a short note on each one, showing damaged structures and the work of reconstruction.

Much of the work was carried out under the direction and according to the design of the service specially set up under the name of Management of New Works.

E. M.

[**625** .23 (42]

HAMILTON ELLIS. — **Nineteenth Century Railway Carriages.** — One volume (7 × 10 1/4 in.) of 176 pages, copiously illustrated. — 1949, London, W.C. 2, Modern Transport Publishing Company, Norman House, 105/109, Strand.

The authors, whose pens have been inspired by railway rolling stock, have devoted their attention of preference to the locomotive. Undoubtedly, its function as the mover, its mysterious shapes and its proportions evoking the idea of power explain the effect produced upon the imagination.

But looked at at closer range, carriages also have their attraction and are a fecond subject upon which to meditate. It is in the atmosphere that he breathes therein that the passenger becomes aware of the quality of the services offered. The more or less generous proportions, the arrangement, the fittings, the decoration, all reflect the customs and mentality of the period. To give the history of passenger carriages is also to narrate the progress of the railway and evoke the political and social evolution of the countrv. And was it not the rolling stock which in developing dictated the growth of the locomotive by requiring of it ever increasing power and speed ?

Many sources of information have been consulted by the author. Ancient railway centres provided him with a wealth of documents. He also consulted the archives of many of the great industrial firms. The paper « Engineering » supplied him with blocks unobtainable elsewhere.

These notes enlighten us regarding the most characteristic arrangement of the most noteworthy carriages used on the English railways from the opening of the Liverpool and Manchester Railway until the last years of the nineteenth century.

The author naturally begins with the body and its arrangement, which is bound to be the main preoccupation at all times since this is the point of view of the passenger for whom the railway exists.

The shape, size, arrangement of seats, interior covering, and accessories have been the subject of much research and undergone many modifications. The author has not overlooked however the technical aspect, such as the construction of the body, and construction and arrangement of the wheels, the suspension, the coupling, the brakes, and lighting. We see the arrival of bogies, inter-communication between carriages, corridor coaches, saloon-cars, sleeping cars. And so during the century, but especially during the second half, the carriage gradually evolved towards its present form.

The human aspect and the pleasant side of things are not overlooked by the author. Often a mere enouncement of facts or laconic description suffices to bring it out. But sometimes, an anecdote is used to illustrate an out of date regulation, a revolution in habits, an old custom, or an innovation in the method of lighting.

He finds an unique occasion in the « Parliamentary carriages » contracted to the « Parlies ». These derive their name from an Act of Parliament (1844) ordering the running every day of a passenger train for third class passengers at the rate of a penny a mile. The « Penny-miler » enjoyed a standard of comfort in these carriages which the reader will appreciate from the details given. He can compare them with the « fourths », which were attached in some cases to a few trains to attract back to the railway the traffic lost to the parallel water services.

In the end, however, the thirds had their revenge, thanks above all to the perseverance of James Allport, who became General Manager and then director of the Midland. They soon occupied a place in this Company of which the author gives an example, an example which was soon imitated. The final result was the dethronement of the seconds which is still a matter of fact.

The illustrations to this book are as interesting as varied. Many of the drawings have never been published before. In addition to the plans and photographs of carriages, sometimes enlivened by the presence of passengers, there are engravings of scenes which occurred at memorable events.

E. M.

[385 (06]

The International Railway Information Office (B.D.C.)

The International Railway Union, which has its headquarters offices in Paris, at 10, rue de Prony, whose membership includes the great majority of European Railways, has decided to create an *International Railway Information Office (B. D. C.)*.

This decision is bound to be of the greatest interest to the railway world at a time when staff questions, financial and legal questions, and problems concerning economics, co-ordination and technical progress must be carefully followed and understood.

The management of this new Office will be confided to an appointed Administration for four years.

The S. N. C. F. has accepted the management of the Office for the first four year period. Its specialised staff, its offices, its collections of books and periodicals, its equipment and its photocopying and microfilming laboratory will be put at the disposal of the new organisation.

The International Railway Information Office will work within the framework of the U. I. C. and with the collaboration of other Administrations whose representatives will form a small Committee. The member Administrations, affiliated Administrations and Administrations sharing in the U. I. C. will also be members of the B.D.C.; in addition other railway administrations, international organisations, and public transport undertakings and firms whose work is bound up essentially with that of the railway will be allowed to belong.

The new Office should be extremely valuable: it bears witness to the close technical collaboration that exists between the European Railways.

Full details (membership of the B. D. C.; subscriptions, etc.) can be obtained from the Manager, Bureau International de Documentation des Chemins de fer, 27, rue de Londres, Paris (IX°).

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P. GHILAIN,

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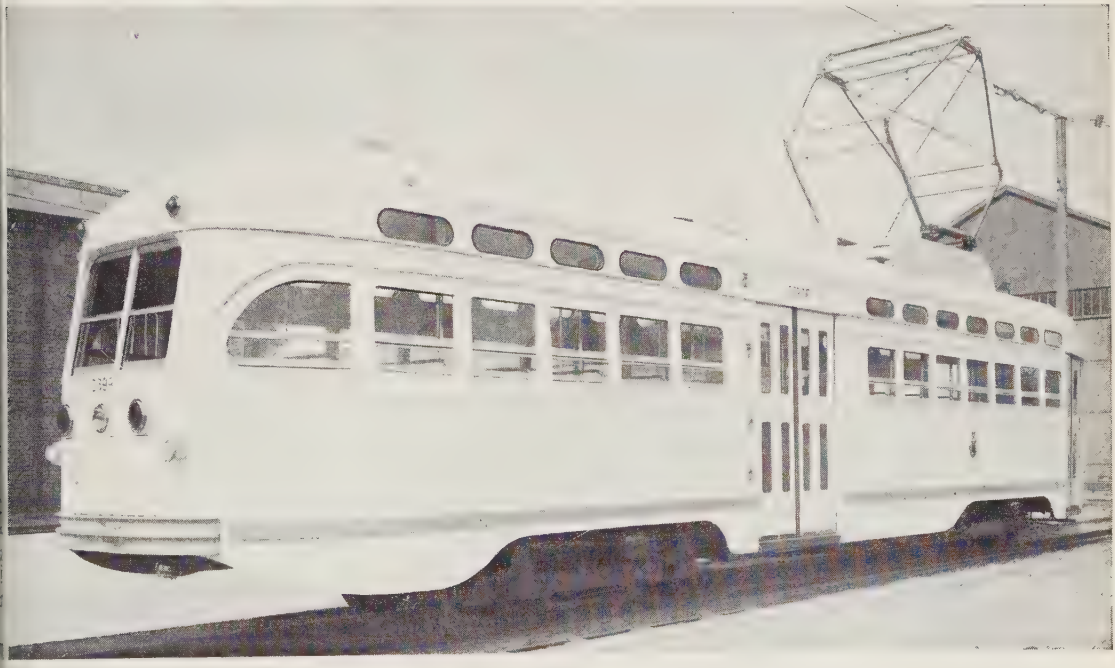

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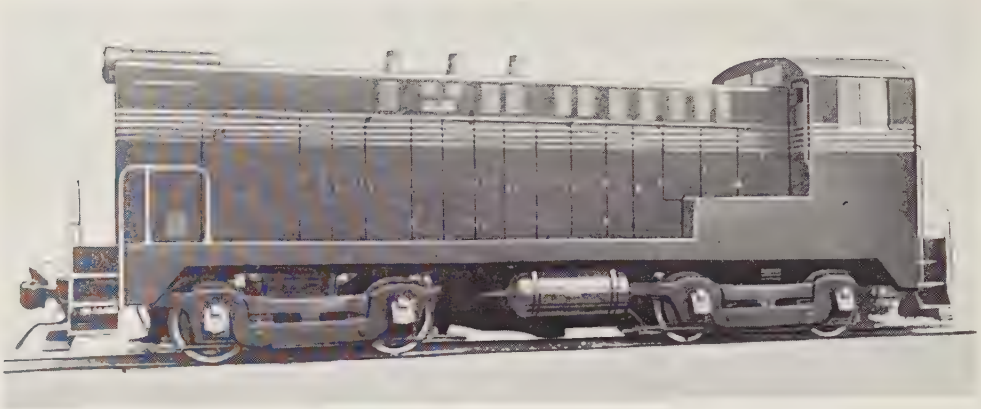

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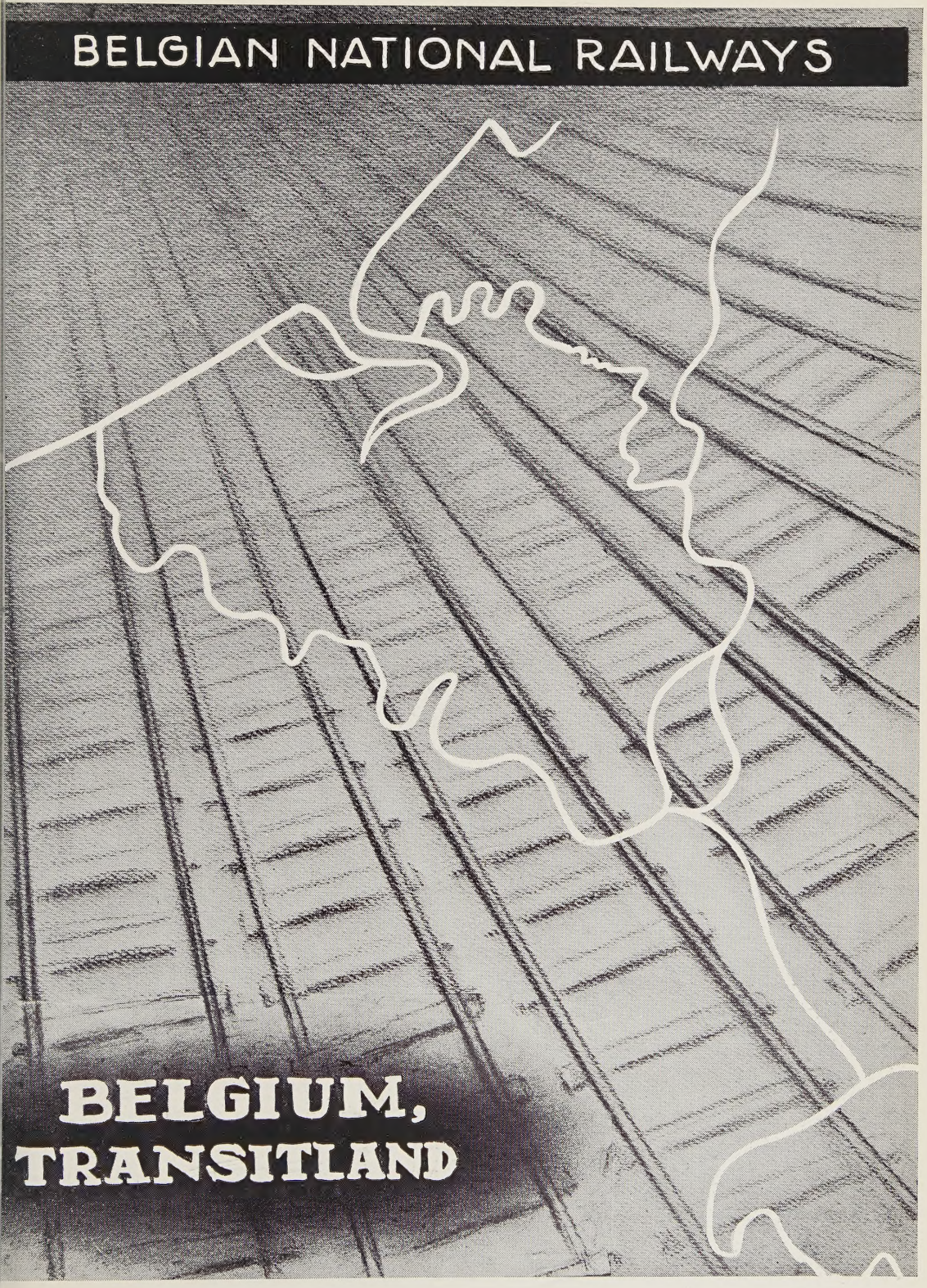
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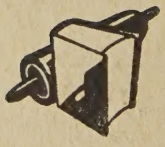

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